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Contract No. NAS8-5407 Report No. BPAD-864-15479

Bendix - BPAD - 864 - 15479

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ELECTROPNEUMATIC LINEAR GIMBAL ACTUATION SYSTEM

MODEL NV-B1

REPORT PERIOD

29 JUNE 1963 THROUGH 29 JUNE 1964

1 August 1964

Submitted to

National Aeronautics and Space Administration
George C. Marshall Space Flight Center
Huntsville, Alabama

By

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ABSTRACT

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This report covers the design effort and component and systems tests performed from 29 June 1963 to 29 June 1964 on two Model NV-B1 Pneumatic Actuators for thrust vector control of the J-2 Rocket Engine. These actuators were designed to conform to MSFC Specification 50M35003.

The prime intent of this program was to prove the feasibility of an electropneumatic linear actuator for thrust vector control. The actuator was designed with all possible component safety factors to insure satisfactory operation; therefore, size and weight specifications were considered secondary for this prototype design.

The design, development and tests proved feasibility of the use of an electro-pneumatic actuator for thrust vector control.

The testing reported herein indicates that improved performance of the design is attainable by further development of various components. Size and weight requirements of the specification may also be satisfied using the basic design proven under this program.

Author

SECTION I

INTRODUCTION

1.1 PROGRAM OBJECTIVE

The objectives of this program were to design, fabricate, and develop two electro-pneumatic actuators to prove feasibility of this type actuator for thrust vector control for the J-2 engine as it would be used on the SIVB stage of the Saturn vehicle.

The actuators were designed to meet the requirements of MSFC Specification 50M35003 which included the following environments:

Temperature:	-250° to 150°F
Pressure:	10^{-6} mm Hg. to sea level
Vibration:	40 G's

The units were to be capable of seventy-two hours of operation. The pneumatic power medium was gaseous hydrogen at 700 to 850 psig at -250°F. The forces required demanded an output power rating of approximately six horsepower.

1.2 PROBLEM AREAS

It was recognized that the prime problem area would be concerned with lubrication at the -250°F temperature. This temperature indicated that dry film type lubrication would be necessary for gears, bearings, and the ball screw. Previous experience had indicated that conventional methods of dry film lubrication of ball screws were not satisfactory. The debris formed from MoS2, etc. tends to cause the balls to slide. A component test program was required to develop a new method for ball screw lubrication.

Cryogenic seals for reciprocating motion were also considered as a problem, since little was known of seals that would seal low pressure hydrogen over the temperature range. A component test program was also established for this program.

Development of a servo valve and motor for operation in the required environment was also necessary. A two-stage servo valve which is similar to previous valves which Bendix has developed for cryogenic applications was designed. A double lobe vane motor was selected over other approaches as a result of computer and design studies which indicated it to be most applicable in comparison to other designs.

1.3 REQUIREMENTS

In addition to the environmental and operating medium requirements, the actuator was required to meet the following design specifications:

a.	System Bandwidth	8 cps
b.	Phase Lag at 1 cps	20 degrees
c.	Peak Amplitude Ratio	1.5
d.	Weight	75 pounds
e.	Duty Cycle (1/2 Rated Load, 1/2 Stroke, 1/2 cps)	72 hours
f.	Quiescent Flow	0.5 lb/min.
g.	Stroke	2.9 ± .032 inches
h.	Snubbing Stroke Each End	0.100 inch
i.	Maximum Load	42,000 pounds
j.	Rated Load	23,200 pounds
	Make Up 7300 in. lbs. friction	1,400 slugs
k.	Rated Velocity @ 23,200 lbs.	1.66 in./sec.
l.	Acceleration	21 in./sec.^{2}
m.	No Load Velocity - Maximum	3.3 in./sec.
n.	Actuator Stiffness - Structural	600,000 lbs./in.
o.	Side Load Applied at Center of Gravity	5,000 pounds
p.	Rated Servo Current	40-50 ma
q.	Servo Valve Power	250 milliwatts
r.	Threshold	0.25 ma
s.	Null Bias Current	0.50 ma

SECTION II

DEVELOPMENT APPROACH AND ACTUATOR DESIGN

2.1 DESIGN CRITERIA

The prime design criteria was to design an actuator to meet all power and life requirements to prove feasibility of an electropneumatic actuator for thrust vector control of the J-2 engine. Size and weight were considered factors secondary to obtaining reliable long life hardware for this prototype unit. The prototype design was considered as being a device on which endurance testing could be run to establish component capabilities for future design activity.

2.2 DESCRIPTION OF ACTUATOR CONFIGURATION

The Model NV-B1 actuator designed during this program is shown schematically in Figure 2-1. Figure 2-2 shows the complete assembly and Figure 2-3 is an exploded view. The parts list, 2775024, for the actuator is included in Appendix A.

Correcting signals to the torque motor of the spool type servo valve controls the hydrogen flow to the high speed vane motor. The motor output drives through a planetary gear train and an idler gear to rotate the nut of the ball screw assembly. Rotation of the nut causes the ball screw to move linearly; rotation of the screw is prevented by a ball spline assembly. The position of the screw is fed back to the system servo amplifier by an internally mounted potentiometer. Actual and desired output positions are compared electrically; any resulting error is modified and amplified and then fed to the torque motor.

The actuator is equipped with snubber springs which act as shock absorbers to prevent structural damage to the actuator and engine.

2.3 DESCRIPTION OF MAJOR COMPONENTS

2.3.1 D. C. Servo Amplifier

The Model 9513 is a special purpose D. C. amplifier assembly designed to sum three separate, ground referenced input signals and drive a floating two wire permanent magnet torque motor. The assembly consists of a wired rack adapter containing a Model 9552 Voltage Amplifier, a Model 9553 Feedback Board, a Model 9554 Power Booster, and a Dual Power Supply.

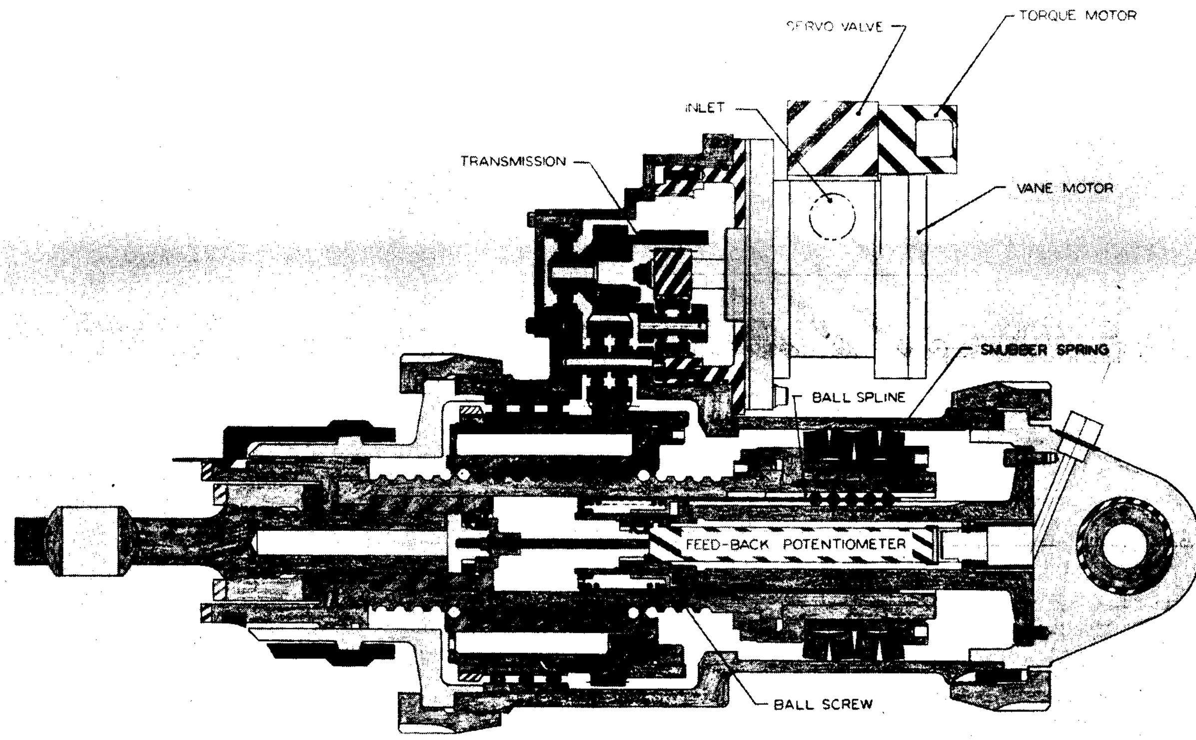


Figure 2-1. Schematic Diagram of the NV-B1 Electropneumatic Linear Gimbal Actuator

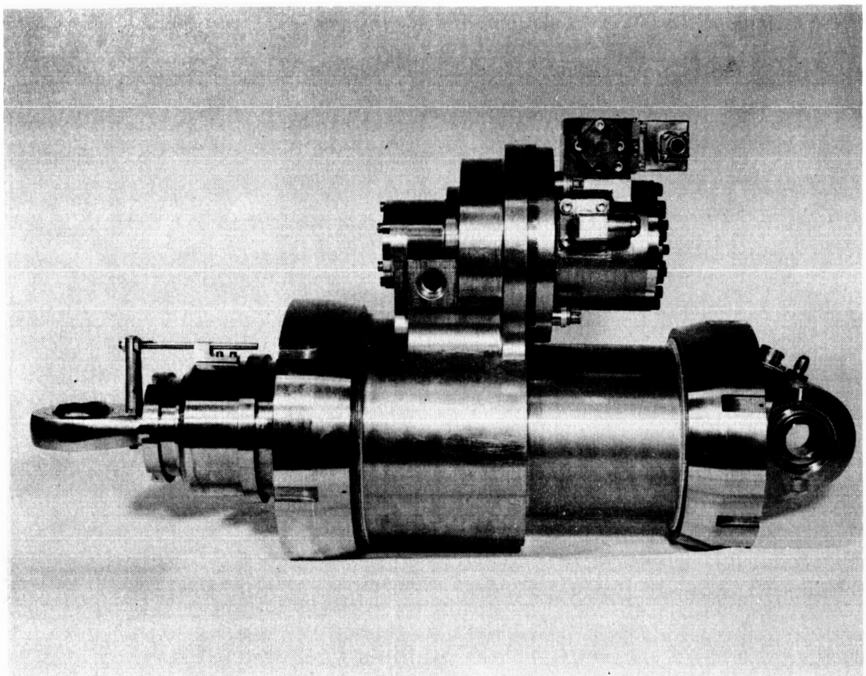


Figure 2-2. External View of the NV-B1 Actuator

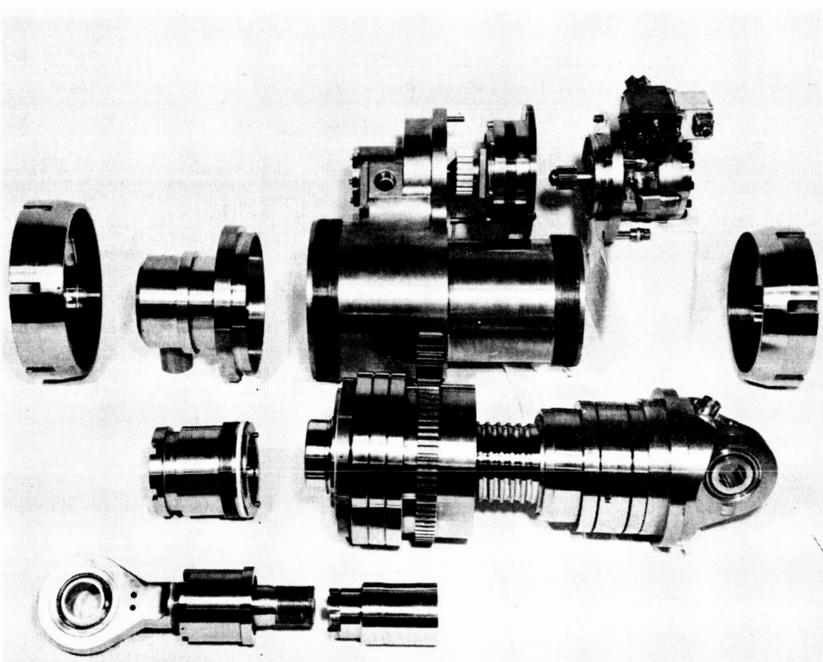


Figure 2-3. Exploded View of the NV-B1 Actuator

The amplifier is designed to provide a flat frequency response to 150 cps and with minor internal connections, a double lead and a double lag for system compensation. The shipping configuration was with the compensation in.

2.3.2 Torque Motor

The torque motor used to actuate the servo valve is a permanent magnet type motor actuated by a ±50 milliampere signal applied to a 100 ohm equivalent resistance coil. The torque motor is supplied by Midwestern Instruments, Inc. as Model 39-4 and by Servotronics, Inc. as Model 29-2-32. Current applied to the coil causes the flapper output to move, controlling the flow in the pilot stage of the servo valve which, in turn, controls flow to the pneumatic motor.

Maximum torque motor stroke is ±0.020 inch but the torque motor adjustable armature stops are set to limit spool stroke to ±0.015 inch. Input power for a 0.020 inch stroke is 0.250 watt maximum, developing three ounces of force, at the end of stroke.

The air gaps and coils are sealed from the working fluid to prevent contamination and to better control gas leakage paths. Leakage of non-toxic or non-explosive gases is normally not a serious problem but excessive leakage of hydrogen to an environment containing oxygen, especially during systems checkout, could be dangerous. Isolation of the air gaps and coils is accomplished by the construction features of the flapper-spring tube assembly. A teflon coated metallic seal is used to seal the valve body/spring tube base interface.

2.3.3 Servo Valve Description

The servo valve is a two-stage pneumatic four-way spool valve actuated by a 250 milliwatt signal applied to a dry coil type torque motor. A ±50 milliampere current signal displaces the 0.625 inch diameter spool ±0.015 inch in a direction dependent upon applied current polarity. The spool supply metering area is 0.0186 in.^{2} and the exhaust metering area is 0.0294 in.^{2} at rated stroke. The valve is designed to operate on cold gaseous hydrogen at -250°F and 800 psia. The valve body and spool material is 440C stainless steel with a black oxidized surface per MIL-13924A, Class 2. The valve is shown in Figures 2-4 and 2-5. A schematic of the servo valve is shown in Figure 2-6.

The schematic diagram, Figure 2-6, shows the internal mechanism of the servo valve. The nozzles of the pilot stage are symmetrically located on either side of the torque motor flapper. Each nozzle is communicated to the opposite spool end or ram chamber. Flow into the ram chamber is through a fixed upstream orifice which, in this valve, is the annular opening created by the radial clearance of the spool in the valve body bore. This pilot stage arrangement requires no filter and employs no small channels in the valve body for pilot stage communication. As in conventional servo valves, an input current signal applied to the torque motor results in a flapper displacement which creates a pressure differential across the spool due to unbalanced flow conditions. The unbalanced force on the spool is in such a direction that spool motion restores the

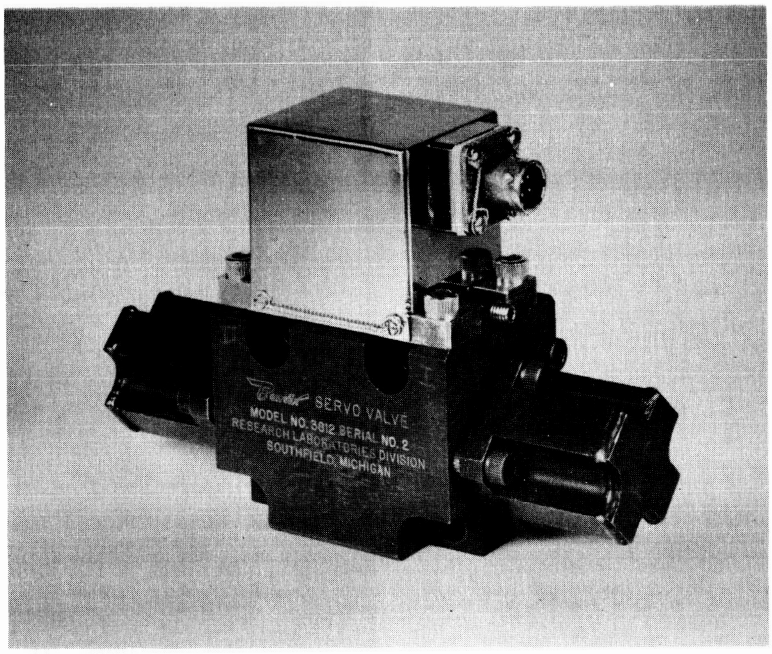


Figure 2-4. Bendix Servo Valve

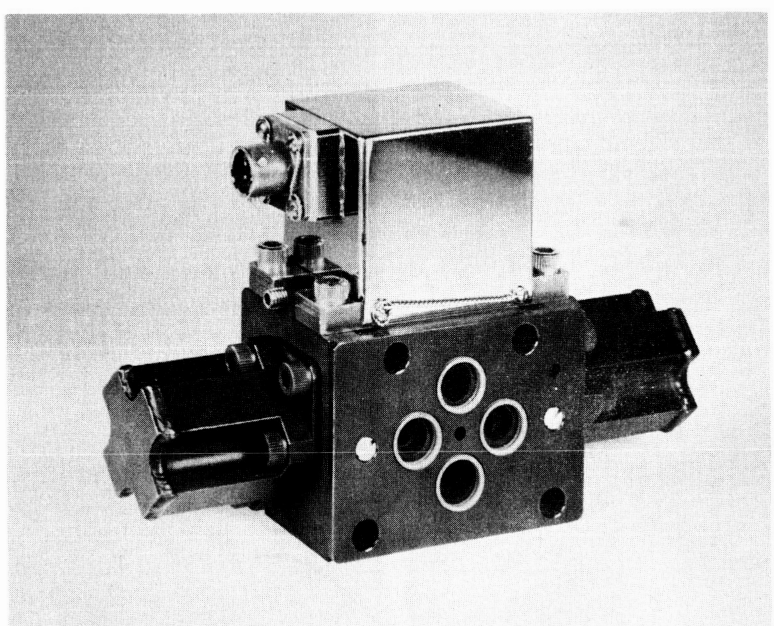


Figure 2-5. View of Servo Valve Showing Ports

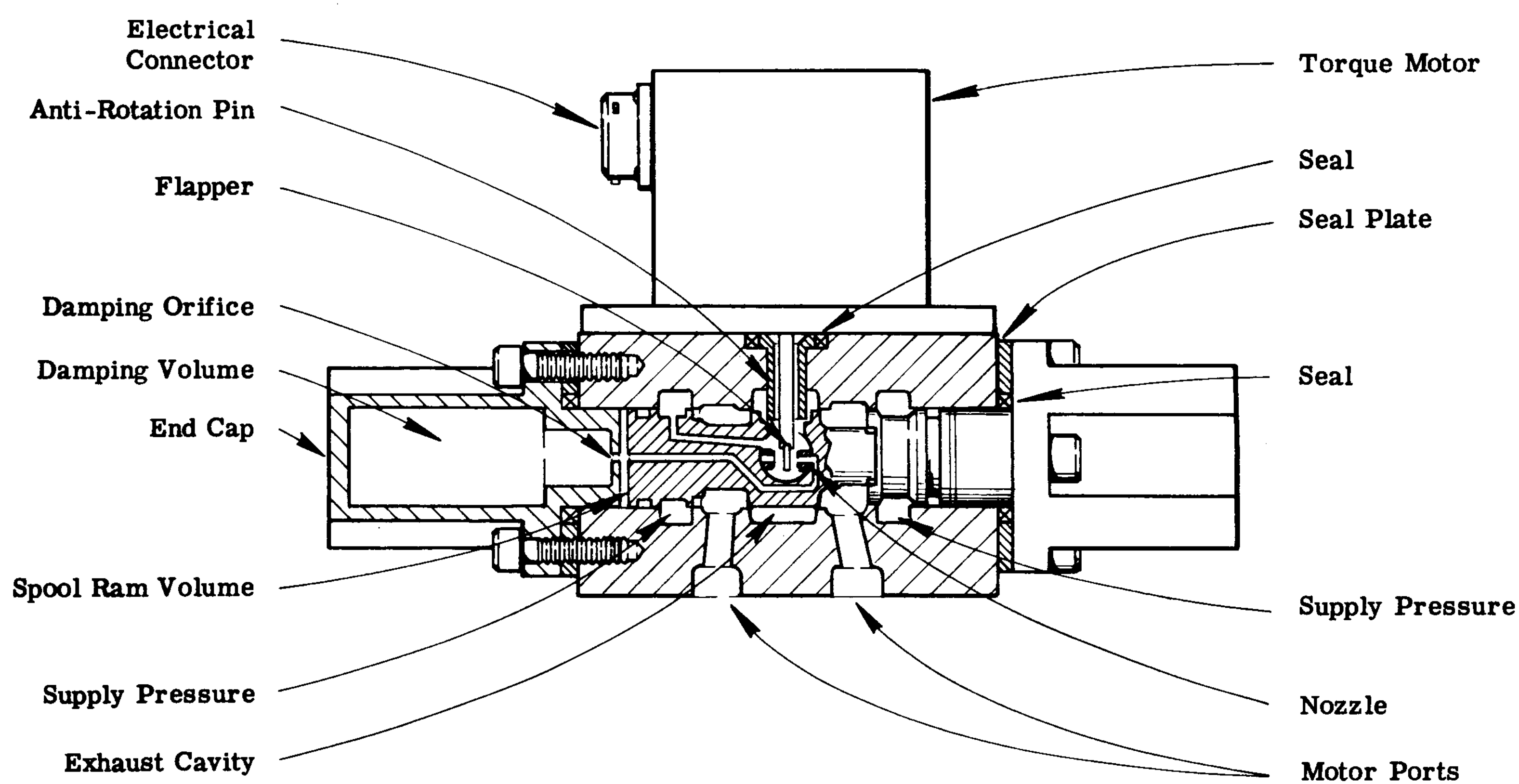


Figure 2-6. Schematic Diagram of Servo Valve

flapper-nozzle assembly to the original centered condition. The spool has been displaced an amount identical to the flapper motion. This one-to-one position feedback enables the spool to be accurately positioned because of the large force available to overcome spool inertial, flow, frictional, and contamination shearing forces.

When the spool is moved from the null position, one load port is opened to supply line pressure and the other to the exhaust line. Since the spool follows the flapper motion almost exactly, the valve metering area is directly proportional to the input signal to the torque motor.

A problem inherent in a pneumatic servo valve is the dynamic instability of the spool that results from gas compressibility. The damping tank connected to the ram chamber of the spool through the small orifice provides sufficient damping for stability on air, nitrogen, or hydrogen. As the spool oscillates, energy is dissipated, pumping the ram chamber gas through the damping tank orifice. This energy loss results in the spool motion decaying to essentially zero amplitude in a very short time. The tank and the orifice have been sized to provide stability throughout the temperature and supply pressure range of the valve. The tank is contained in the end cap for minimum over-all valve size.

Lubrication of the spool is provided by the oxide film on both the body and spool sliding surfaces.

2.3.4 Servo Motor Description

The servo motor is a double lobe positive displacement vane motor and consists of a housing, rotor, vanes, vane lift springs, rotor support bearings and two end caps. An exploded view of the breadboard model is shown in Figure 2-7. An assembled view of the motor used on the actuator is shown in Figure 2-8 with servo valve and torque motor.

The rotor is located in the center of an elliptical cam bored in the housing. When rotating, nine vanes slide on the cam contour while reciprocating in slots machined in the rotor. Springs are inserted under the vanes to insure vane contact with the cam contour when the motor is not rotating. The motor is completely reversible and used in a positive displacement mode of operation.

In a positive displacement design, the working gas is exhausted without expansion and work derived from the motor is displacement energy. Essentially, the unit will receive gas at a certain pressure and temperature and discharges the gas at close to the same conditions.

Gas under high pressure is metered from the valve to the inlet port to the chamber formed by two adjacent vanes, the rotor, the housing and the two end caps. The pressure difference across the vanes then causes a force unbalance and rotation begins. As each vane chamber rotates into position it is filled with high pressure gas and positive displacement energy is obtained. This charging process continues until the trailing vane of that chamber cuts off the inlet port, then to insure against "short circuiting" an

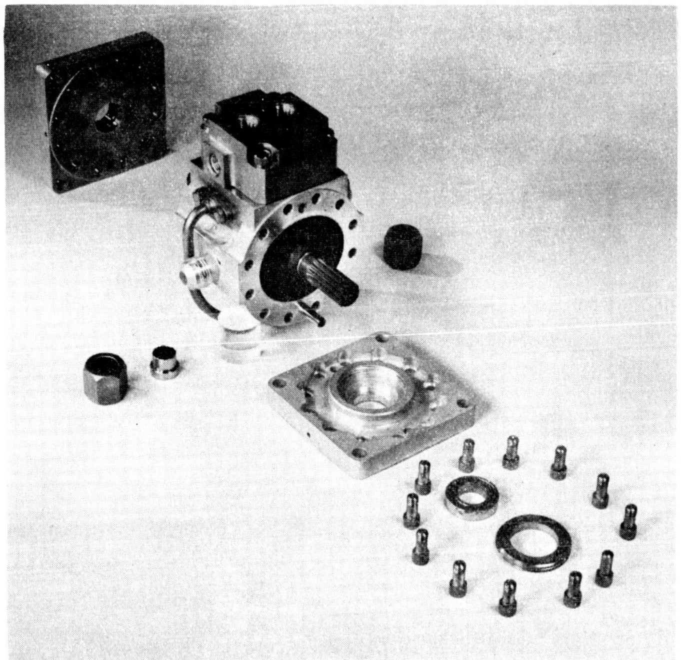


Figure 2-7. Exploded View of Vane Motor Assembly

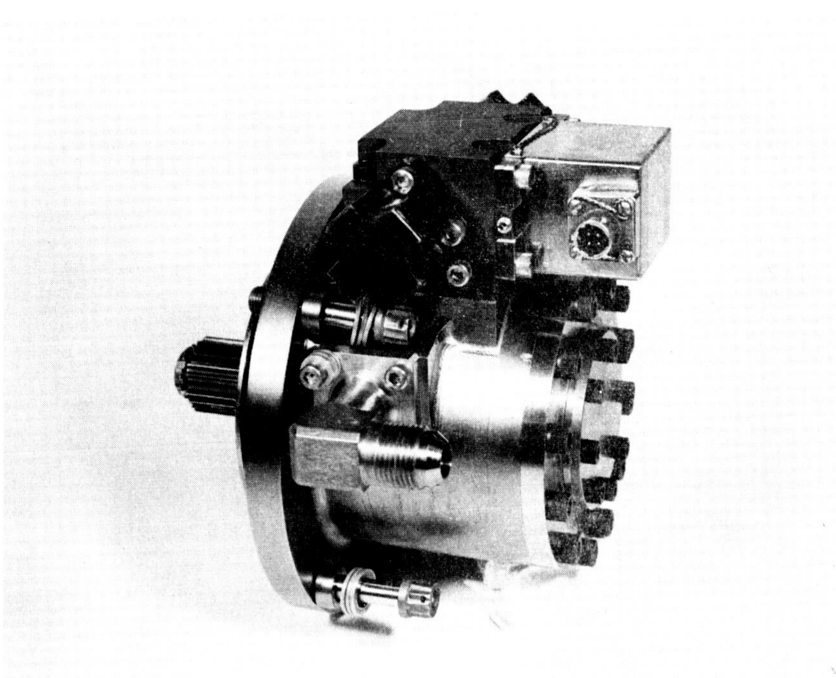


Figure 2-8. Servo Motor Assembly

extremely small expansion cycle begins. As the leading vane of the chamber uncovers the exhaust port, this design safety margin expansion is terminated, gas flows to the ambient at near initial pressures and the final pressure in the chamber is ambient or a controlled back pressure. After further rotation, the trailing vane "cuts off" the exhaust and the residual or trapped volume of gas is ported through the servo valve.

The motor design utilized in the J-2 system is semi-balanced because although it has two lobes, it has an odd number of vanes.

Utilizing a double lobe design tends to balance the forces on the anti-friction rotor support bearings. An odd number of vanes causes slightly higher bearing loads but increases minimum starting torque and reduces output torque ripples. The increase minimum starting torque results from not having a condition where two vanes are in the minimum extended position at the same time.

2.3.5 Transmission

A single stage planetary transmission with a ratio of 4.5:1 is used between the motor and ball screw nut. The output gear drives an idler gear which in turn drives a gear pinned to the ball nut providing an additional ratio of 3.13:1. Figure 2-9 shows the planetary transmission. The total transmission ratio motor to ball screw nut is, therefore, 14.08:1.

The transmission gears are 400 series stainless steel with differential hardness between the gears. Lubrication is provided by coating the planet gears and the idler gear with .0003-.0007 inch thick Hi-T-Lube, a proprietary dry film coating consisting of gold, silver and MoS_2 applied by the General Magnaplate Corporation, Belleville, New Jersey.

Figure 2-10 indicates relative positioning of the gear train to drive the ball screw nut.

2.3.6 Ball Screw and Ball Spline Assembly

The ball screw and ball spline assembly was manufactured by Beaver Precision Products Corporation, Clawson, Michigan. The screw, nut and male splines are made of Latrobe Steels BG-42 CRES bearing steel. Titanium carbide 0.28125 inch diameter balls are used for load carrying in both the screw and spline to reduce the possibility of vacuum welding in space environments. The screw has a lead of 0.400 inch per revolution and has a basic diameter of 3.00 inches. The ball nut consists of three separate circuits of 2-1/2 turns per circuit. Each circuit contains 105 balls. Lubrication is provided by inserting one 0.280 inch diameter teflon ball alternatively with each nine and then ten load carrying balls. This method of lubrication was developed during the component testing phase of this program.

The ball spline consists of six spline ball tracks containing five balls per track. A leaded bronze, spring-centered ball retainer is used to provide proper ball spacing for the spline. The relatively slow rolling speed of the spline balls does not require lubrication.

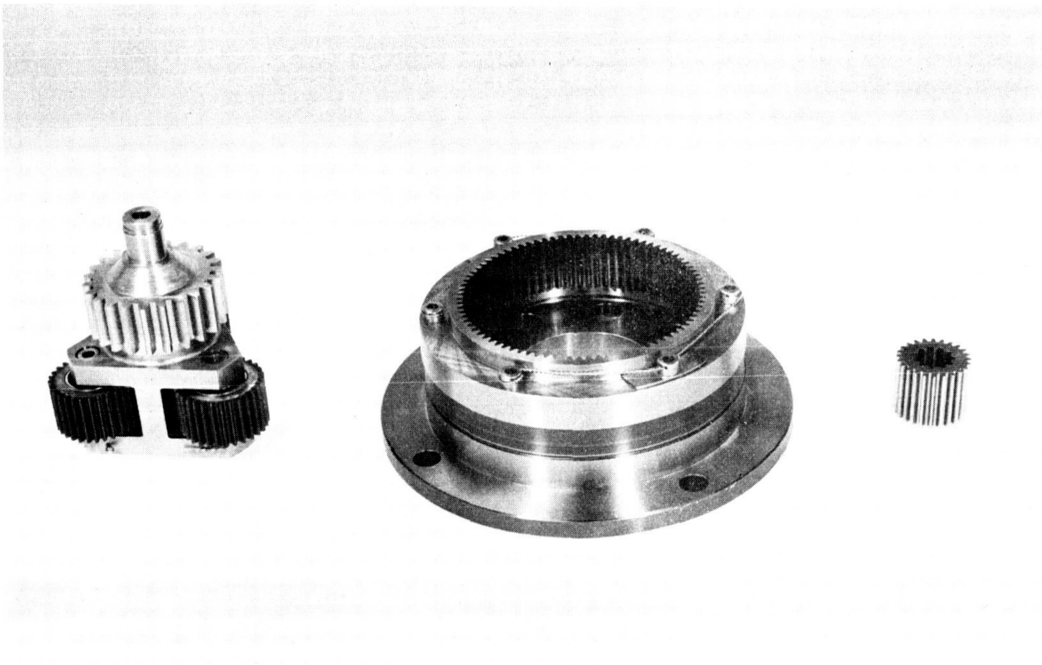


Figure 2-9. Exploded View of Planetary Transmission

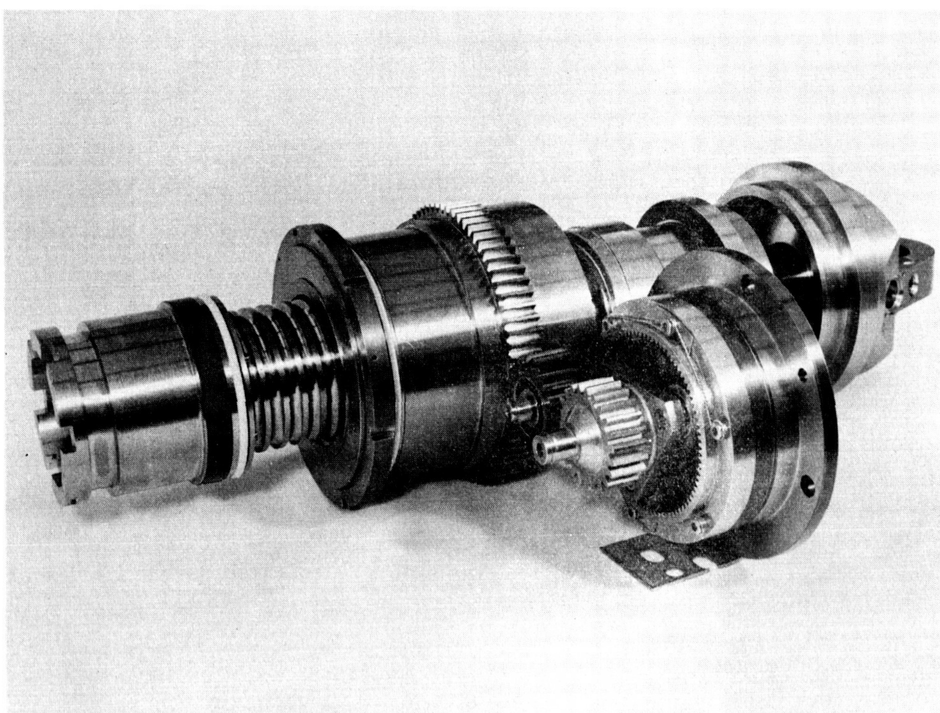


Figure 2-10. Planetary Transmission and Ball Screw Assembly

Figure 2-11 shows the ball screw and ball spline assembly.

2.3.7 Ball Bearings

The ball bearings used in the motor and transmission are the split outer race type and use races and balls of 440C stainless steel. In order to provide lubrication, solid retainers of Du Pont's Super Polymer SP-1 are used. Ball contact with the retainer transfers SP-1 material to the races to provide a lubrication film which has been found to provide good life characteristics for cryogenic applications.

The actuator thrust is absorbed by a specially designed matched set of three four-point contact bearings which can absorb thrust in both directions. SP-1 is also used for ball retainers of the split riveted type for these bearings.

2.3.8 Static Seals

The static seals used were primarily metallic teflon coated face seals except for the split line of the motor and the adjustment of the rod end which were cup type teflon seals with garter spring expanders. The cup type seals were required to provide a diametral seal.

2.3.9 Dynamic Seal

The dynamic seal used on the rod end of the actuator was a cup type seal made of teflon with a garter spring expander. The reciprocating motion requires a diametral seal. The outer lip of this seal works against the chrome plated reciprocating bearing bore.

2.3.10 Reciprocating Bearing

The reciprocating bearing used on the rod end piston to prevent metal-to-metal contact consists of a trapped 0.033 inch thick, 0.560 inch wide strip of Rulon type LD material. The ends of this material are cut on an angle so that the greater thermal contraction of the Rulon will simply widen the gap around the annulus retainer.

2.3.11 Spherical Ball Ends

The spherical ball ends use 440C balls in a formed 17-4 PH outer race and are secured in the rod end and bracket eye (head end) by circular staking on the outer race over a chamfer on the housing.

2.3.12 Housing

The main housing was machined from a solid billet of AMS 5613 CRES steel. The cover on the rod end and bracket eye on the head end are secured in place by large spanner nuts. All highly stressed threads conform to MIL-S-8879 configuration with special root radii to prevent stress concentration.

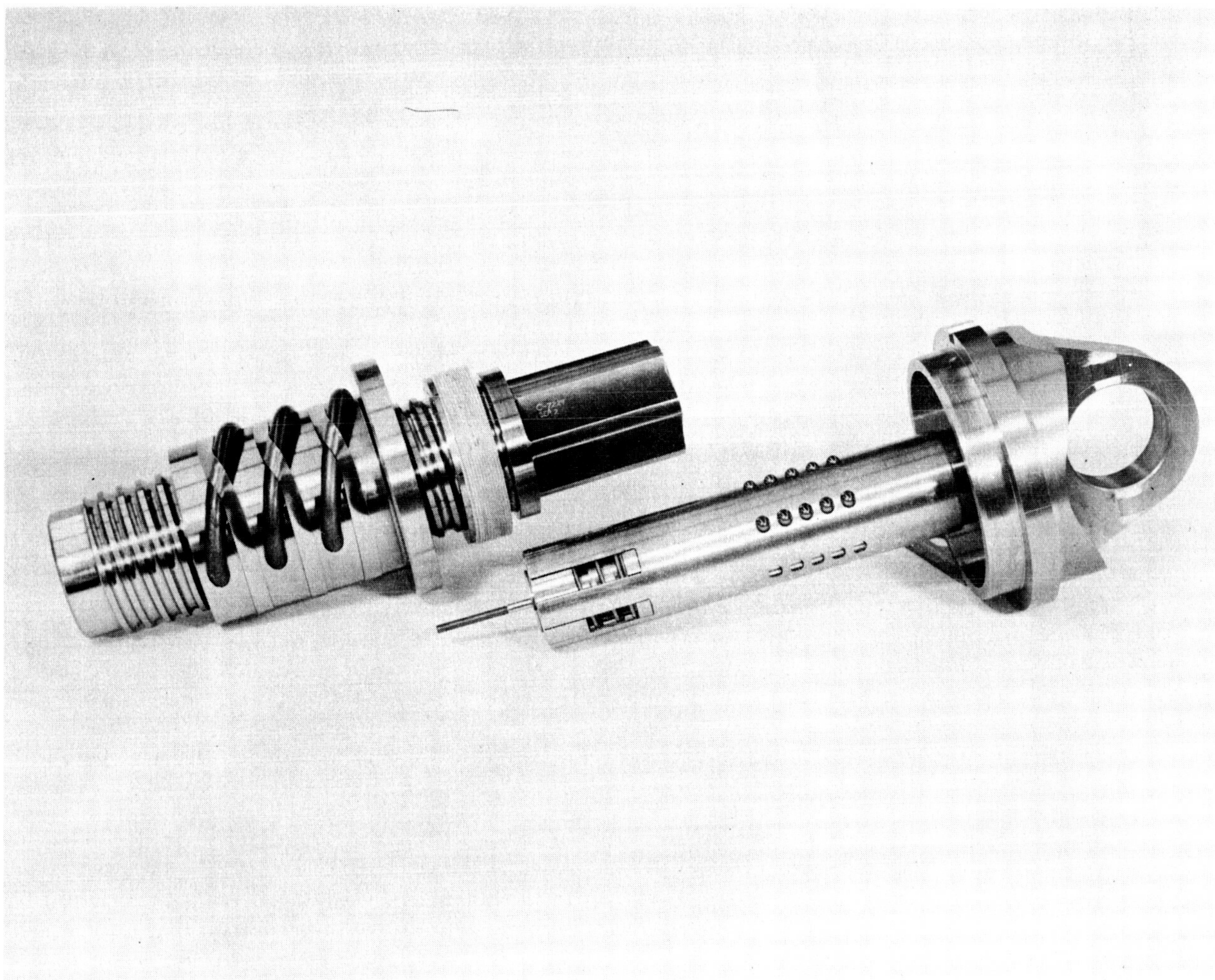


Figure 2-11. Ball Screw and Spline Assembly

2.3.13 Feedback Potentiometer

The feedback potentiometer used in the NV-B1 design conforms to Bendix Drawing 2775565 and Specification NPD-110 (included in Appendix B of this report). This potentiometer is of special design and includes a Bendix Scintilla PT1H-12-8-P receptacle so that it may be plugged into the actuator and can be removed by simply removing the rod end and its two retaining nuts, prong-type locks for the nuts, and the wiper shaft connection. This unit was purchased from the Markite Corporation.

2.3.14 Snubber

The snubber is composed of one assembly of four disc springs arranged in series. To maintain the contact point at the spring's outer diameter when snubbing in either direction, only three springs are active in any one direction. The spring material is AMS 5528 or AMS 5644B (17-7PH plate or forging) heat treated to condition TH 1050. This material was selected for its availability, high stress capability and retention of ductility at cryogenic temperatures.

The spring assembly is mounted on the ball screw and contacts the actuator housing at both ends of the 2.9 inches travel. The available snubber stroke is 0.1 inch, the spring rate is 632,000 lbs./in. providing a maximum force of 63,200 pounds.

SECTION III

DESIGN ANALYSIS

3.1 CONTROL SYSTEM

An analog computer study was made of the control system with a two-stage jet pipe valve instead of with the final valve configuration. The study is conservative in its results because the dynamics of the jet pipe valve is not quite up to those of the valve used. The results of this study are included in Appendix C.

3.2 MOTOR-TRANSMISSION SIZING PARAMETERS

The actuator Specification 50M35003 requires the following output power requirements.

1. Rated load-speed — 23,200 lbs. load at 1.66 in./sec. (5.83 HP)
2. Maximum stall force — 34,800 lbs. subsequently increased to 42,000 lbs. in a MSFC-Bendix coordination meeting on August 6 and 7, 1963
3. Maximum no-load speed — 3.3 in./sec.
4. Bandwidth of 8 cps, for a command signal of 0.05 inches amplitude, with 7300 lbs. load friction, and 116.5 lbs.-sec.^{2}/in. load inertia.

Using these requirements and estimated inertias, the required actuator component characteristics are as follows:

Motor Inertia at Motor	0.002 in.-lb.-sec.^{2}
Planetary Input Inertia at Motor	0.00104 in.-lb.-sec.^{2}
Planetary Output Plus Ball Gear Nut Inertia at Nut	0.1389 in.-lb.-sec.^{2}
Ball Screw Ratio	15.71 rad./in.
Planetary Transmission Efficiency	90%
Ball Screw Efficiency	80%

A study was made assuming various overall ratios of 150 to 250 radians per inch. The results of this analysis are shown in Figure 3-1. It was decided from this analysis that an overall ratio of approximately 225 radians per inch would more nearly match vane motor characteristics than any other ratio. The final design provided a ratio of 221 radians per inch. Figure 3-2 indicates the requirement points and the area in which motor torque-speed characteristics should fall to meet all requirements. This curve incorporates the actual actuator inertia values and overall efficiency. It was noted that the normal straight line torque speed curve of a pneumatic motor must be modified to meet all requirements. It appeared that some method of limiting maximum motor speed would be the best method of meeting all requirements.

3.3 SERVO VALVE AND TORQUE MOTOR

3.3.1 Pilot Stage

A schematic and equivalent pneumatic circuit of the pilot stage of the servo valve selected is shown in Figure 3-3. This type of bleed arrangement was considered to be the most satisfactory, because for a low temperature transient, the spool with its lower mass would decrease in diameter more than the bore of the servo valve body. Since orifices A_1 and A_2 are clearances, their areas would increase causing a slight decrease in the pressure sensitivity of the pilot stage, whereas if the servo valve configuration shown in Figure 3-4 were used, the increase in clearance would cause more leakage and would result in a lower ram pressure and a slight loss in pilot stage control.

A study was made of the pilot stage configurations shown in Figures 3-3 and 3-4 to determine the most desirable orifice sizes. Early in the study, the configuration shown in Figure 3-4 was eliminated in favor of the configuration shown in Figure 3-3. During the initial phases of the study, orifice A_c was omitted, but was later added when it was found that the pilot stage was marginally stable when the spool to servo valve body bore clearance was made a minimum compatible with smooth spool movement at low temperatures.

Figure 3-5 is a plot of the pilot stage blocked spool ram pressure as a function of the torque motor flapper displacement for various values of orifice A_c diameters with the equivalent diameters of orifices A_1, A_2, A_3, and A_n held constant. The diameter of orifice A_c was chosen to be 0. 016 inch. The constant orifice areas above were selected with due consideration given to the servo valve quiescent flow requirements, the stiffness of the pilot stage, the clearances necessary for cold temperature operation, and the effects of the mass of the spool on the servo valve frequency response.

3.3.2 Power Stage

The power stage of the servo valve is a 0. 625 inch diameter spool with a supply metering area of 0. 0186 square inches and an exhaust area of 0. 0294 square inch at a rated spool stroke of ±0. 015 inch.

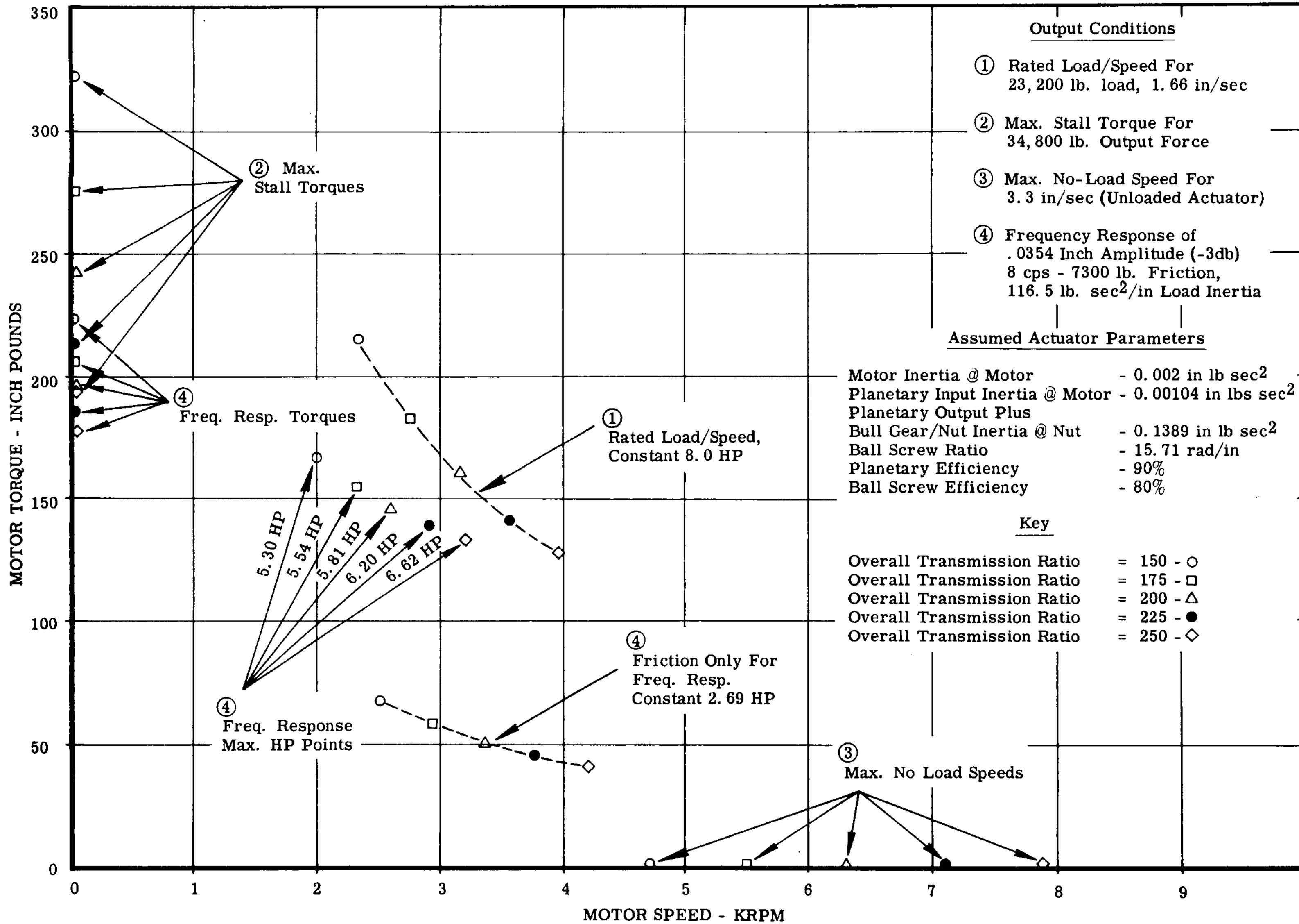


Figure 3-1. Motor-Transmission Sizing Parameters

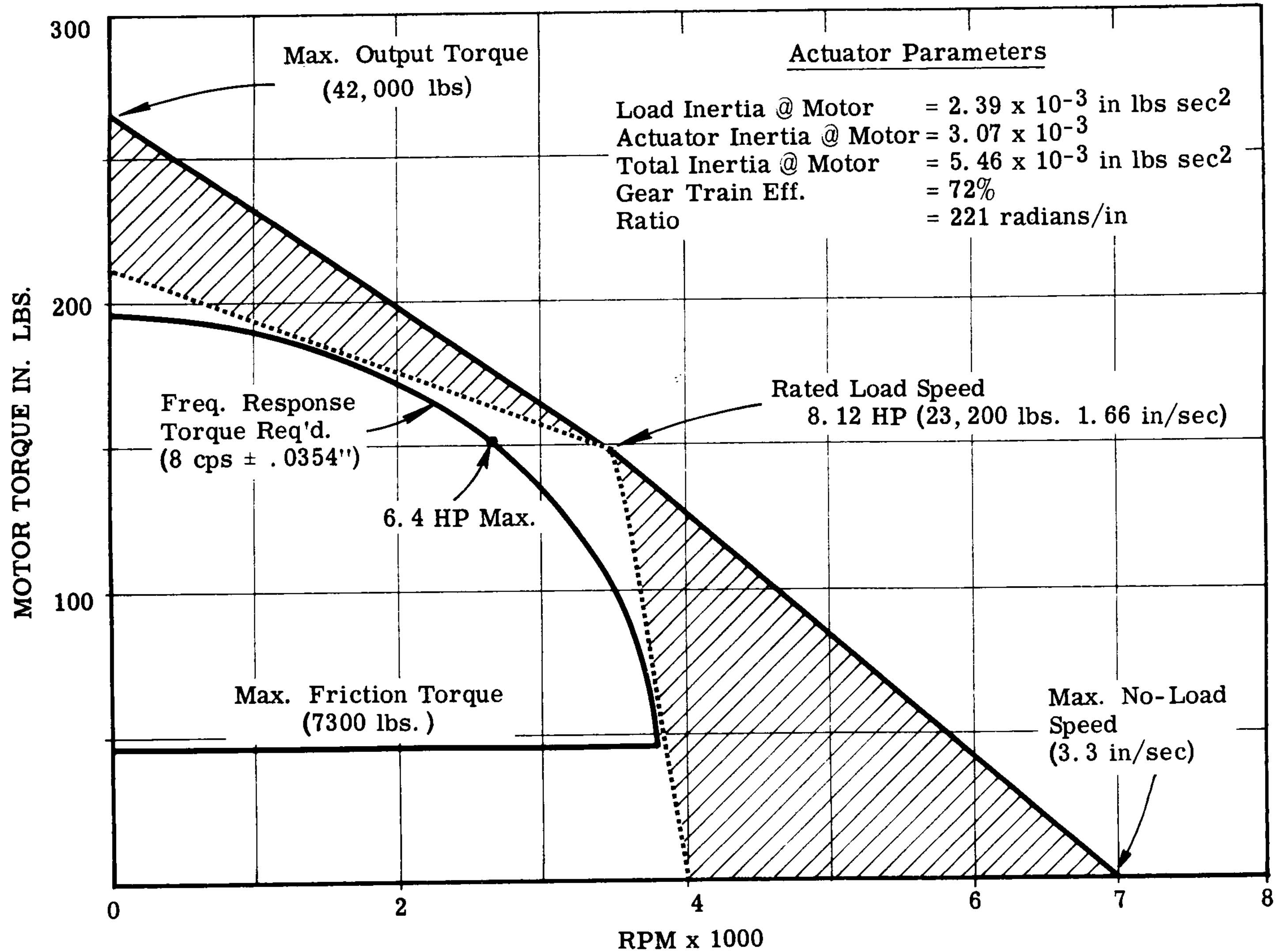
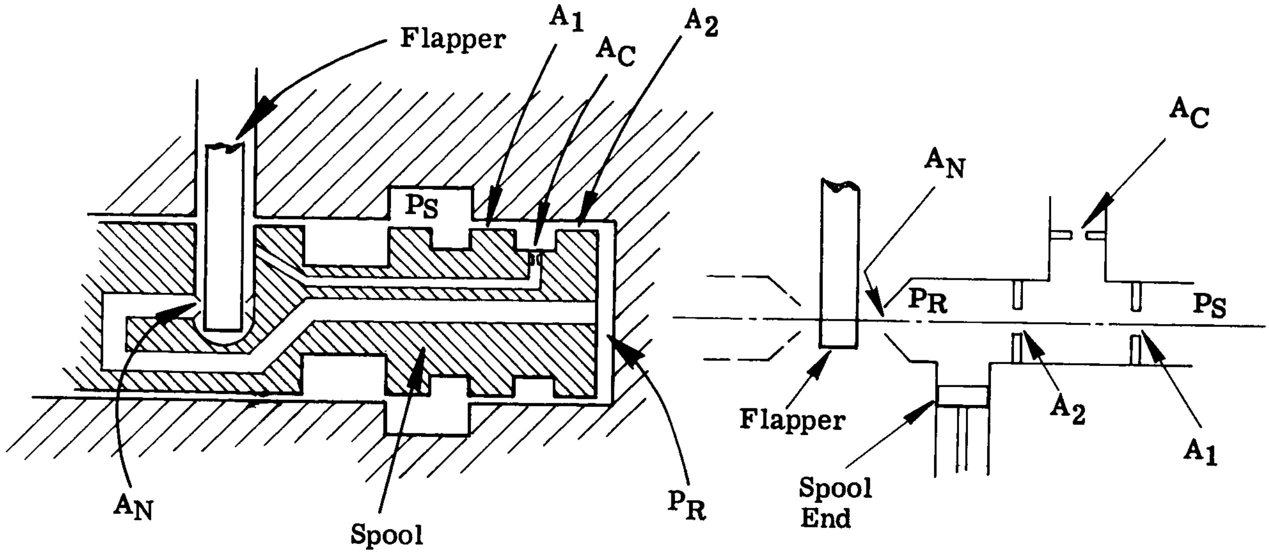


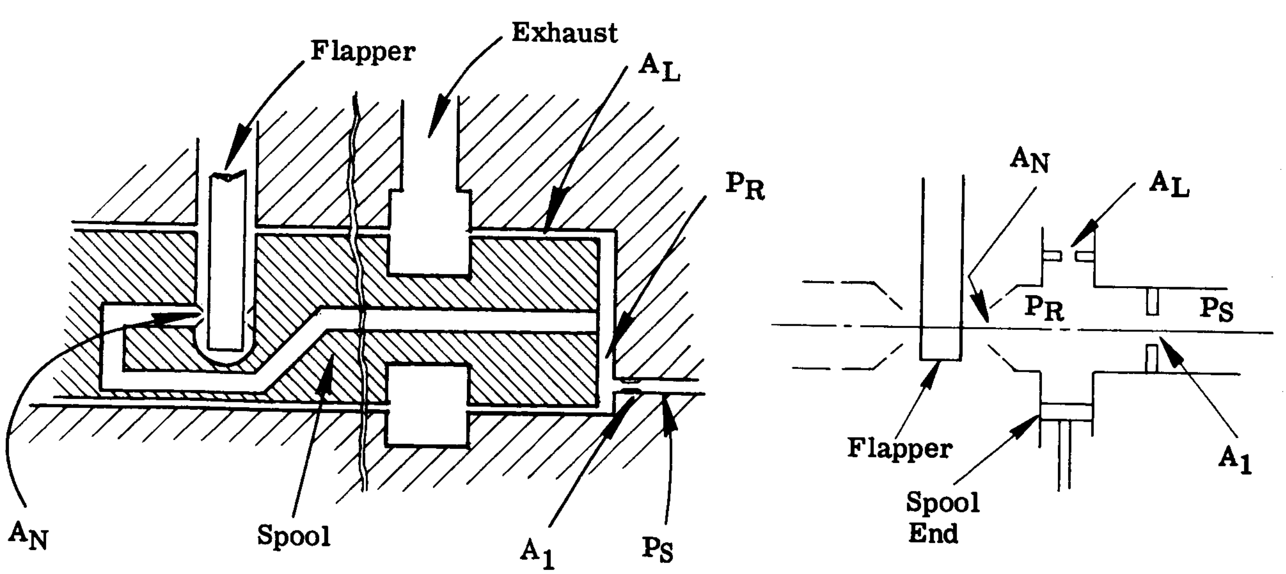
Figure 3-2. Motor Torque Vs. Speed Requirements



(a) Schematic - Pilot Stage

(b) Equivalent Pneumatic Circuit

Figure 3-3. Pilot Stage Configuration



(a) Schematic - Pilot Stage

(b) Equivalent Pneumatic Circuit

Figure 3-4. Alternate Pilot Stage Configuration

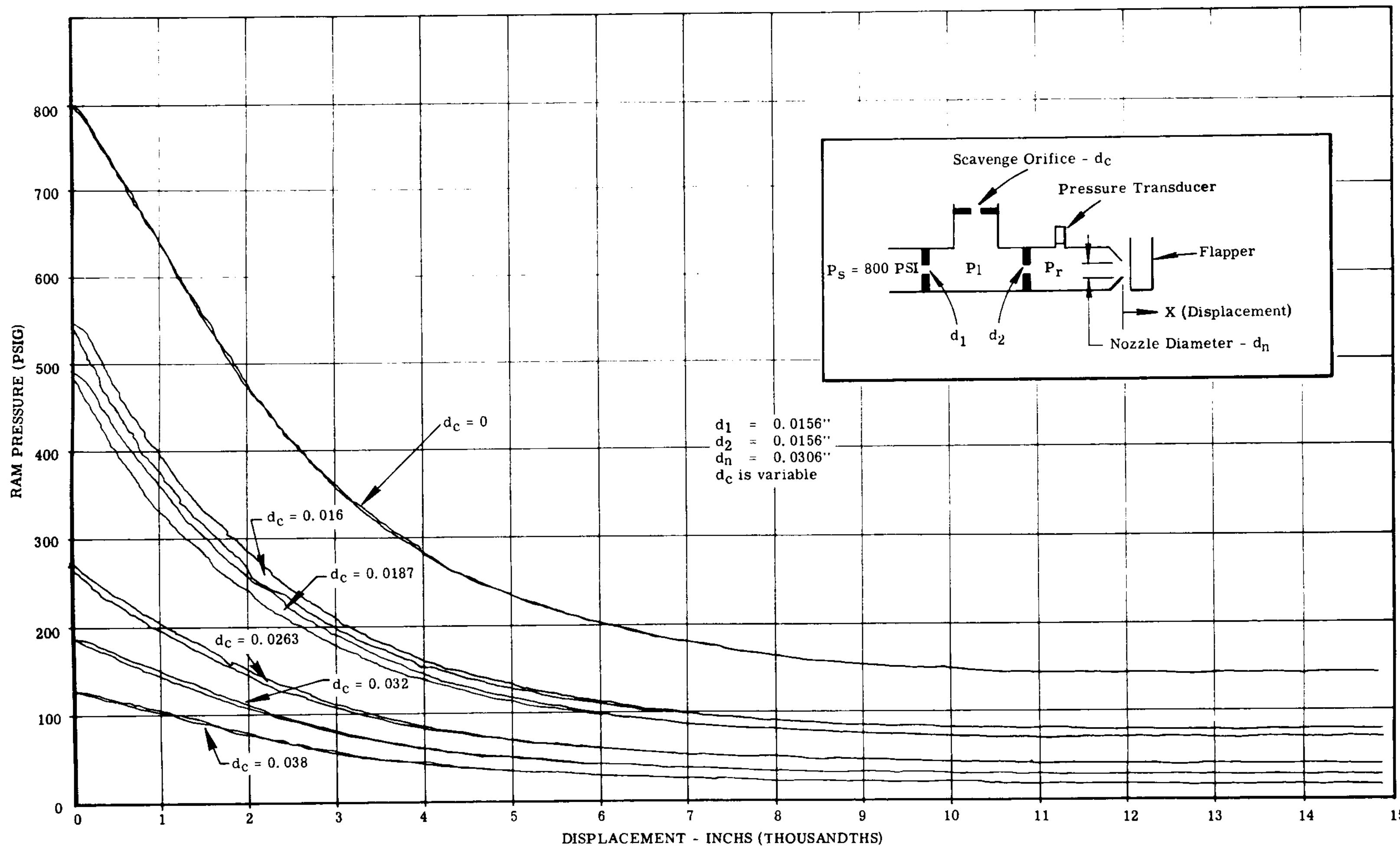


Figure 3-5. Pilot Stage Ram Pressure Sensitivity with Blocked Spool

The above metering areas could have been obtained with a spool diameter in the order of 3/16 to 1/4 inch. The larger diameter of 0.625 inch was chosen to minimize the problem of mounting the flapper nozzles in the spool.

The exhaust area was made larger than the supply area to minimize the exhaust back pressure on the motor. This was accomplished with a single diameter spool by utilizing the full circumference of the spool for the exhaust area and only a portion of circumference for the supply area. The supply area consists of three equally spaced 76° angular openings.

Should it be desirable to increase the supply area, the angular openings can be increased from the present 228° to the full 360° which would result in the supply area being equal to the exhaust area. A further increase to 0.039 square inch can be obtained by increasing the spool stroke to ±0.020 inch.

3.3.3 Material Selection

The material selected for the valve spool and body was 440C stainless steel. This material has been found to be very suitable for valve applications. Tests previously conducted at Bendix Research Laboratories Division confirmed that wear resistance, corrosion resistance, and material stability made 440C stainless acceptable for low temperature valve applications. Additional factors considered in the selection of 440C was the availability of the material, relative manufacturing advantages (i.e., no exotic processes required), and the ability to retain an oxide film on the surface.

The J-2 actuator must be run without conventional lubrication. Initially it was believed that a 440C spool could operate in a 440C bore without galling, but during preliminary valve tests very slight gall marks were noticed on the spool lands. To alleviate this, a black oxide film was applied to the spool and bore to serve as a lubricant between these surfaces. The black oxide also provides another advantage in that the material becomes even more corrosion resistant. In the many hours of operation after applying the oxide treatment to the three valves, no further galling was observed.

A temperature stabilization technique was applied to the critical valve parts made of 440C material. The stabilization consisted of cycling the valve parts, before final fitting, several times through a temperature range greater than the expected operational temperature range. This technique minimizes subsequent valve distortion due to temperature extremes.

3.3.4 Valve Seals

To prevent hydrogen gas leakage to atmosphere, teflon coated metallic seals are used rather than the conventional noncoated metallic seals. The teflon coating is desirable because of its cold flow properties under load which allows the teflon to flow and fill any minute surface irregularities. Thus far the seals appear satisfactory.

3.3.5 Contamination Tolerance

The valve can, within reason, be considered a contamination tolerant valve because of the large pilot stage shearing force capability, no critical small orifices, and a circular annulus serving as the upstream pilot stage supply orifice instead of a small conventional orifice. All orifices were made as large as was compatible with the quiescent flow requirements to allow large particles to pass without clogging. A circular annulus was chosen for the upstream orifices because it will accept a larger number of contaminate particles without clogging than will a circular orifice. A large particle, which would normally block a conventional orifice, blocks only a small angular segment of an annulus permitting flow to continue through the remainder of the annulus.

3.3.6 Torque Motor

The torque motor requirements were not too severe except for the low power which requires a somewhat larger torque motor, and also limits the maximum flapper stiffness. Normally, because of servo valve stability considerations, the flapper should be as stiff as possible. To increase the flapper stroke, the output point of the flapper was extended to a position one inch below the base of the torque motor.

The valve body size was designed so that pressure feedback could be incorporated, if necessary, at some future date. This factor caused the body size to be slightly larger than would be otherwise necessary.

A design feature is the shielding effect provided by the anti-rotation tube. The anti-rotation tube projects into the spool to prevent the spool from rotating into the flapper. The tube shields the flapper which passes through the tube from pneumatic "noise" or turbulence as gas flow is exhausted through the valve.

A torque motor centering adjustment is provided on the valve. By adjusting two set screws, in opposition to one another, it is possible to center the torque motor at a point which provides equal valve motor port pressures.

3.4 ANALYSIS OF PNEUMATIC VANE MOTOR

3.4.1 Displacement Volume (D_m)

In a non-expansion motor, the volume of gas which is the only contributor to the useful output work is the displacement volume, D_m. The volume V_m is defined as the useful volume of gas for one chamber between any two vanes. Physically, V_m, is the total volume (V_T) between two vanes at admission cut-off, minus the clearance volume ($V_c + V_v$).

$$V_m = V_T - (V_c + V_v)$$

D_m, required to determine torque and output work is given as the displacement volume per radian of output or

$$D_m = \frac{V_m \, n \, N}{2\pi}$$

where n = total number of vanes
and N = number of lobes in cam contour

Clearance Volume: $(V_c + V_v)$

$(V_c + V_v)$ is the volume of gas between the trailing vane and the outside diameter of the rotor and the cam contour when a vane just uncovers the admission inlet port plus the volume of gas under and around the leading vane. Little or no positive displacement is gained by the motor from this volume. It could be argued that the volume of gas under the vanes, during admission inlet, is doing work in moving the vane out of the slot, and in keeping the tips in contact with the cam contour by overcoming slot frictional forces. Also, if a differential pressure existed across the extended vane area, the gas under the vane would force the vane against the housing and thus result in an added reaction torque on the rotor resulting theoretically, in output work. It is desirable for this undervane pressure to insure vane-tip-to-cam contact, however, if the cam-contour-to-tip reaction force is high, the torque gained from this condition would be overwhelmingly exceeded by the tip frictional losses. Ideally the vane should "just touch" the cam contour.

On the basis of the above paragraph, no practical positive-displacement output energy can be derived from the clearance volume.

Since, in the non-expansion motor, energy is completely or nearly completely derived from displacement energy, the gas charged into the clearance volume, unable to contribute displacement energy, is reflected as a fuel loss. Essentially, a void is being charged and discharged and the gas is only being pumped at a sacrifice of fuel. Therefore, a prime design consideration is to keep the clearance volume as small as practicable.

3.4.2 Sizing the Motor

In the initial sizing of the motor for torque, consideration must be given to the load friction, inertia, and acceleration as stated in the motor design specification.

Simulated Load Inertia	4.1×10^{-3} lb-in-sec^2
Maximum Motor Inertia	1.1×10^{-3} lb-in-sec^2
Load Frictional Torque	47 in-lbs
Load Acceleration (for response)	19.8×10^3 rad/sec^2

A general expression for the required motor torque is as follows:

$$T_m = \left(J_m + \frac{J_1}{e_1} + \frac{J_2}{e_1 e_2} + \ldots + \frac{J_n}{e_1 e_2 \ldots e_n}\right)\alpha_m + \frac{T_L}{e_1 e_2 \ldots e_n}$$

T_m = Maximum total torque output of the motor required (in-lb)

J_m = Inertia of the rotor (in-lbs-sec^2)

J_1, J_2, J_3 = Reflected inertia to the motor of the various components between the motor and the load (in-lbs-sec^2)

α_m = Angular acceleration in rad/sec^2

J_ℓ = Load inertia reflected to the motor $= \left(\frac{J_1}{e_1} + \frac{J_2}{e_1 e_2} + \ldots \frac{J_n}{e_1 e_2 \ldots e_n}\right)$

T_L = Frictional load

e_1, e_2, e_3 = Efficiencies of various components downstream of the motor.

If the overall efficiency of the transmission is assumed to be 72%,

$$T_m = \left(1.1 + \frac{4.1}{.72}\right) \times 10^{-3} \text{ in-lb-sec}^2 \times 19.8 \times 10^3 \frac{\text{rad}}{\text{sec}^2} + \frac{47 \text{ in-lbs}}{.72}$$

T_m = 134.5 in-lb + 65.2 in-lb

T_m = 199.7 in-lb (Required motor stall torque)

The displacement (D, in^3/rev) of the motor required is expressed by

$$D = \frac{2\pi \quad T_m}{\Delta P \times E_T}$$

where:

E_T is the torque efficiency (80% based on previous experience with vane motors)

ΔP = Differential pressure out of the valve at full load (500 psi)

$$D = \frac{2\pi \times 199.7 \text{ in-lbs}}{500 \text{ psi} \times .80}$$

$$D = 3.2 \text{ in}^3/\text{revolution}$$

3.4.3 Vane Stress Levels and Deflections

are

Vane material is Du Pont SP Polymer. The basic vane dimensions are:

thickness	= t	=	.140 inch
height	= h	=	.625 inch
length	= ℓ	=	2.625 inches
maximum vane extension =	V_E	=	.125 inch

with 850 psi on the vane in the maximum vane extension position, the following resulting conditions occur:

maximum torque producing force = $F_T = V_E \ell (\text{Pressure}) = 280$ lbs

maximum reaction force in slot of rotor = $F_{R1} = \dfrac{F_T V_E}{2(h - V_E)} = 35$ lbs.

maximum reaction force at O. D. of rotor = $F_{R2} = F_T + F_{R1} = 315$ lbs

maximum static bending stress on vane = $\sigma_B = \dfrac{MC}{I} = \dfrac{F_{R1}(h - V_E)\,6}{t^2 \ell} = 2040$ psi

maximum static shear stress (at O. D. of rotor) = $\sigma_S = \dfrac{F_{R2}}{t\ell} = 860$ psi

maximum deflection (.300 inch from tip radius and using S. P. material = d $= \dfrac{F_{R2} b^3}{3EIc} = \left[\dfrac{a(1+b)}{3}\right]^{3/2} = .0015$ inch

where a = .0500 inch
b = .0625 inches
c = .5625

S. P. Characteristics

Tensile strength room temperature	= 13, 000 ± 2000 psi
Modulus of elasticity	= 430, 000 psi (R. T.)
Flexural strength	= 14, 700 psi (R. T.)
Shear strength	= 11, 400 psi
Maximum compressive strength	= 27, 500 psi

3.4.4 Vane Tip Bearing Stress

The Hertz stress or specific bearing pressure at the vane tip can be determined from the following equation

$$\text{Max. } S_c = .798 \sqrt{\frac{P_t\left(\frac{D_h - D_v}{D_h D_v}\right)}{\frac{1 - r_h^2}{E_h} + \frac{1 - r_v^2}{E_v}}}$$

where:

S_c = Specific bearing pressure, psi

P_t = Vane tip load-in pounds per linear inch

D_h = Diameter of curvature of cam contour, inches

D_v = Equivalent vane tip diameter, inches

r_h = Poisson's Ratio - housing

E_h = Modulus of Elasticity of housing, psi

r_v = Poisson's Ratio - vanes

E_v = Modulus of elasticity - vanes

The radius of curvature of the cam is continuously changing, the point of maximum centrifugal force and maximum pressure force occur near the minimum radius of curvature on the cam. The worst condition occurs at the 90° point where the centrifugal force is a minimum, and the pressure force nearly a maximum, and the cam contour radius of curvature a maximum. The worst condition has been examined.

$$P_{t_{max}} = P_1 t$$

where P_1 = Motor Pressure

t = vane slot width

$$\text{Max } S_c = .798 \sqrt{\frac{P_1 t \left(\dfrac{D_h - D_v}{D_h \times D_v}\right)}{\left[\dfrac{1 - r_h^2}{E_h} + \dfrac{1 - r_v^2}{E_v}\right]}}$$

P_1 = 0 to 850 psi

t = .156 inch

D_h = 2.064 inch minimum, 2.730 inch maximum

D_v = .625 inch .625

E_h = 30×10^6 psi

r_h = .3

E_v = 4.3×10^5 psi

r_v = .3 assumed

$S_c = 238 \sqrt{P_1}$, if P_1 = 900 psi

$S_{c_{max}}$ = 7150 psi on S. P. Vanes

This is a safe factor as compared to a compressive strength of 27,500 psi.

3.4.5 General Equations for the Cam Contour and Centrifugal Forces

The polar equation for the cam contour in a double lobe motor is:

$$\rho = \frac{a\,b}{\sqrt{a^2 \sin^2 \theta + b^2 \cos^2 \theta}}.$$

where

ρ = radial distance from centerline of rotor to point on housing

a = major radius of ellipse

b = minor radius of ellipse

θ = angle at the point in question

The in-slot velocity is given by

$$V_s = \rho^3 \ \omega \ \text{Sin} \ \theta \ \text{Cos} \ \theta \ \frac{(b^2 - a^2)}{a^2 \ b^2}$$

The cam contours relative acceleration or deceleration is

$$a_n = \frac{\omega^2 \ (a^2 - b^2)}{a^2 \ b^2} \left[\left(\frac{3}{2} \ \rho^2 \ \frac{V_s}{\omega} + \rho^3 \alpha \right) \text{Sin} \ 2 \ \theta + \rho^3 \ \text{Cos} \ 2 \ \theta \right]$$

where

α = acceleration of rotor (rad/sec^2)

For all practical purposes

b = r_r (rotor radius)

a = $r_r + 2 \ e$

Solutions to the general equations after numerical substitutions are made are plotted in Figures 3-6 and 3-7.

The predominant force, disregarding pressure forces, is the centrifugal force acting on the vane. In either design the centrifugal force is considerably larger than the cam contour normal force.

If no points of inflection, (washboarding) exist on either types of cams, no vane bounce can theoretically exist for either design. If vane bounce could exist it would have to be on the discharge portion of the cycle and the in-slot frictional forces would have to be exceptionally high and variable for slip-stick movement required for vane bounce.

3. 5 PLANETARY TRANSMISSION DESIGN

The planetary transmission was designed on the basis of a rated load of 147 inch pounds input, a rated speed of 3500 rpm input, and an overall gear train and ball screw efficiency of 72%. It was desired to maintain the dynamic bending stress below 60, 000 psi under these conditions. Table 3-1 indicates the stall and dynamic forces on the gear teeth, bending stress under dynamic load, and Hertz stress (compressive stress) under dynamic rated loads.

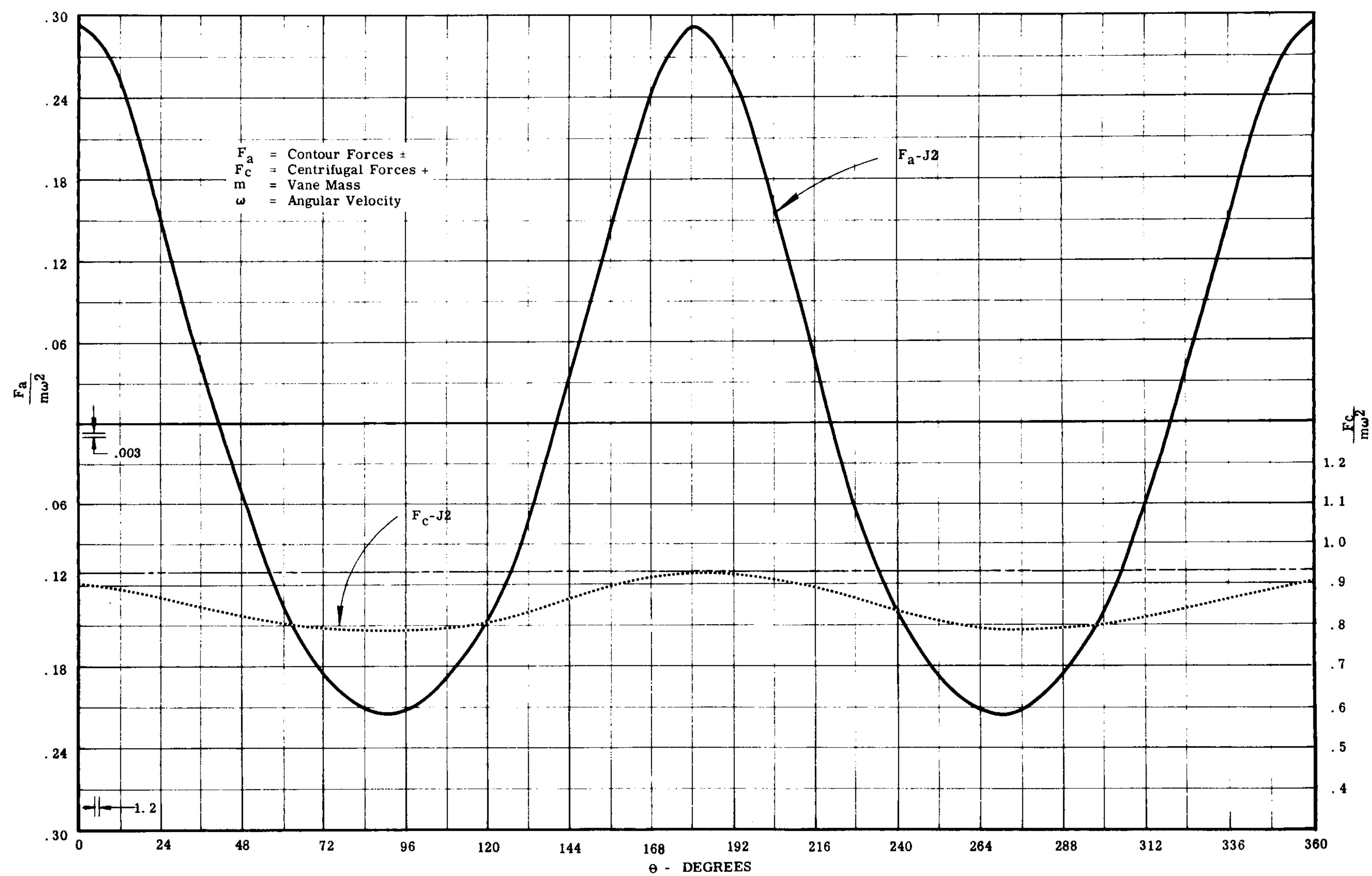


Figure 3-6. Cam Contour Forces Vs. Angular Position

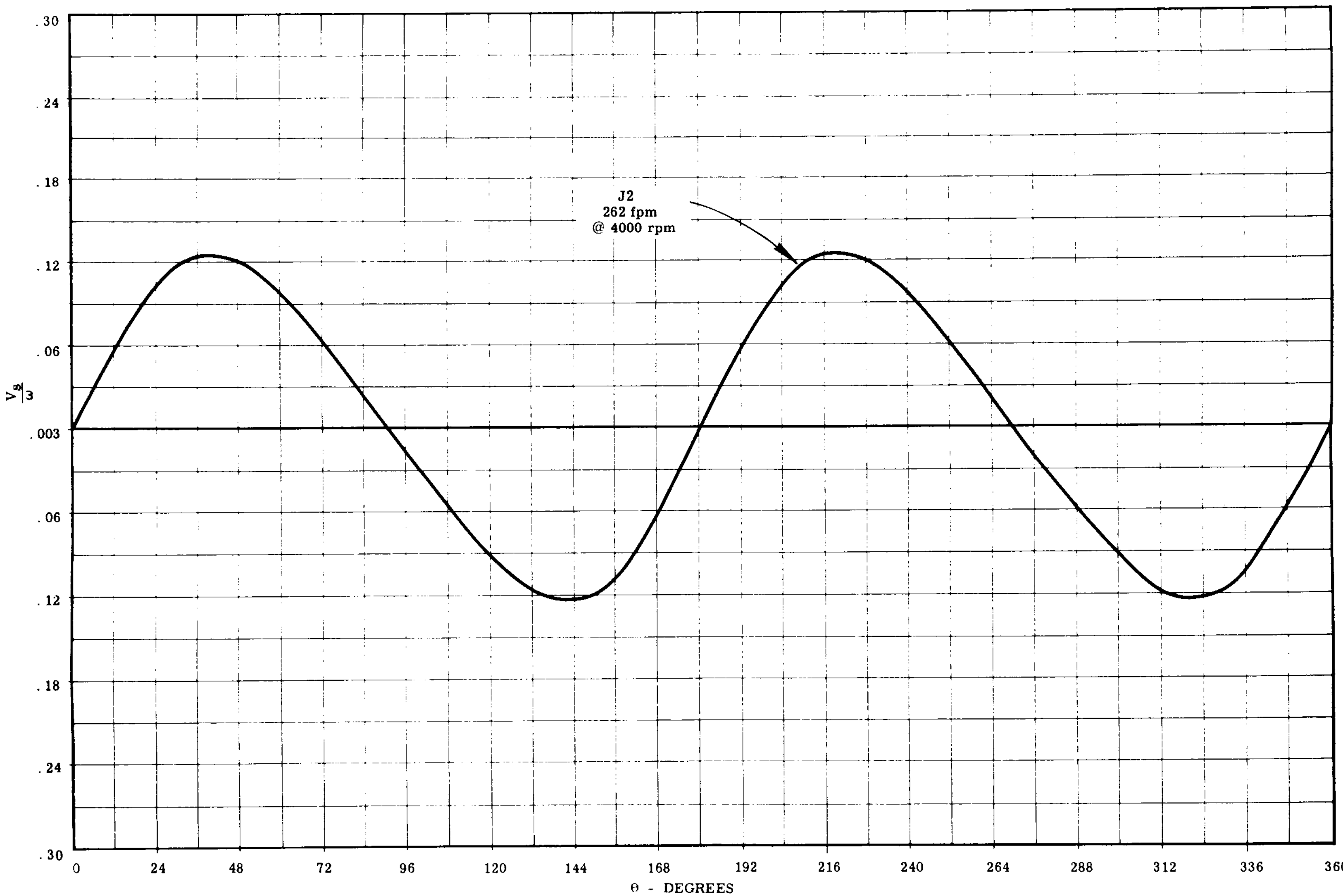


Figure 3-7. "In-Slot" Vane Velocity Vs. Angular Position

Gear	P/N	P. D.	Teeth	Tooth Length	Tooth Load Stall	Tooth Load Dyn. (1)	Stress Bending	Hertz Stress	Hertz Stress
		In.	No.	In.	Lbs.	Lbs.	PSI	PSI Rated	PSI Endurance Load
Sun	2775570	1. 000	24	. 88	175	552	52, 500	198, 000	140, 000
Planet	2775571	1. 250	30	. 75	175	552	49, 200	106, 000	175, 000
Ring Gear	2775600	3. 500	84	. 88	175	552	40, 400		
Output	2775607	1. 917	23	. 75	1240	1175	56, 500	220, 000	155, 000
Idler	2775575	1. 917	23	1. 00	1240	1175	56, 500	168, 000	118, 000
Ball Nut	2775609	6. 000	72	. 75	1240	1175	46, 200		

NOTES:

(1) Dynamic Tooth Loads Considered at Motor Rated Load and Rated Speed (3500 RPM) 72% Eff. and 147 in. lbs. input.

Table 3-1. Planetary Transmission Parameters

It will be noted that the maximum Hertz stress between the planetary output and idler gear is 220,000 psi under rated load. This is the recommended maximum for good lubrication and long life. The endurance test conditions of one-half load will reduce this stress to 155,000 psi which should provide sufficient life with the Hi-T-Lube on the idler gear.

The planet cage in this design is supported by only one bearing on the output shaft and the planet gears against the ring gear at the motor end. This floating of the planet cage permits the planet gears to equally share the motor input torque. The reaction force from the output gear is then shared by the output bearing and the planet gears.

3.6 BALL SCREW AND BALL SPLINE DESIGN

The ball screw was designed to meet the life requirements for the J-2 actuator which is seventy-two hours at 1/2 stroke, 1/2 rated load, at 1/2 cps which corresponds to 1.45 inches, 11,600 pounds for 130,000 cycles at 1/2 cps.

It was determined that a 3.00 inch diameter screw with 0.281 inch diameter balls should be used to carry the above loads. The minimum standard lead of the screw for a 0.281 inch diameter ball is 0.400 inch.

For good lubrication, a 0.281 inch diameter ball is rated at 120 pounds per ball for a life of 10^7 impacts. An impact is defined as the number of times per revolution that a ball will pass one spot on the screw. With a 3.00 inch diameter screw and 0.281 inch ball we have a diameter ratio of 10.6 which will, with the screw being the moving element, result in 16 impacts per revolution.

Total impacts for seventy-two hours (130,000 cycles) were calculated as follows:

$$\text{Cycles} \times \frac{\text{Stroke}}{\text{Lead}} \times \text{Impacts/Revolution} = \text{Total Impacts}$$

$$130{,}000 \times \frac{1.45}{.400} \times 16 = 7.55 \times 10^6 \text{ impacts.}$$

NOTE: Load is reversed, therefore, only one direction is considered.

The load carrying capacity is inversely proportional to the cube root of ratio of impacts.

$$\text{Load/Ball} = \text{Rated Load} \sqrt[3]{\frac{\text{Rated Impacts}}{\text{Actual Impacts}}}$$

$$\text{Load/Ball} = 120 \sqrt[3]{\frac{10^7}{7.55 \times 10^6}} = 132 \text{ pounds}$$

The nut chosen was a standard configuration with three separate circuits with 2. 5 loaded turns per circuit. This would permit the following number of balls to share the load:

$$\frac{\text{Turns} \times \text{Diameter } \pi}{\text{Ball Diameter}} = \text{No. Load Balls}$$

$$\frac{7.5 \text{ turns} \times 3 \text{ in. } \pi}{.281} = 250 \text{ balls}$$

The actual endurance load per ball would be

$$\frac{\text{Endurance Load}}{\text{No. Balls}} = \frac{\text{Endurance Load}}{\text{Ball}}$$

$$\frac{11,600 \text{ lbs.}}{250} = 46.5 \text{ lbs./ball}$$

Prior testing at Bendix using dry film lubricants had indicated that a load safety factor of 4:1 should be used in comparison to well lubricated ball screw design. In this case the safety factor is only $\frac{132}{46.5} = 2.84:1$. This compromise was necessary in order to build an actuator close to the specification envelope. It was therefore necessary to develop a better lubrication method during the course of the program, and a component development program was initiated. See Section 4. 2. 5 for test results.

The ball spline is subjected to an endurance torque as follows:

$$\text{Torque} = \frac{\text{Endurance Load} \times \text{Lead}}{2\pi \times \text{Screw Efficiency}}$$

$$T = \frac{11,600 \text{ lbs.} \times 0.400 \text{ (lead)}}{2\pi \times 80\% \text{ Efficiency}} = 925 \text{ in. lbs. at 1 inch radius}$$

Thirty 0. 281 diameter balls are used and the load per ball is only 30. 8 pounds. In addition, the impacts would only be two per cycle since the balls are spaced at 0. 437 inch and they travel one-half of endurance stroke or 0. 725 inch. The load carrying capacity would be:

$$\text{Load Per Pall} = 120 \sqrt[3]{\frac{10^7}{.26 \times 10^6}} = 338 \text{ pounds per ball}$$

which provides a safety factor of $\frac{338}{30.8} = 11:1$ and can be run with no lubrication.

3.7 BEARING DESIGN

Bendix has for several years conducted programs for the development of cryogenic bearings. One of the later developments is the use of retainers made of Du Pont SP-1 (Super Polymer). Ball action against the retainer transfers a film of SP-1 to the races which serves as a lubricant. This type of bearing retainer used with 440C balls and races was used for all ball bearings.

The radial bearings used in the motor and transmission are of the split outer race type to obtain greatest capacity for size. The criteria used for selecting these bearings was to determine average endurance requirement speed and bearing load based on actuator rated load. The manufacturers' 500-hour minimum or B10 life load rating should equal or exceed the rated load at the bearing average speed. Bendix experience indicates that these bearings should have a life in excess of the 72-hour requirement. Table 3-2 indicates the bearing loads and ratings used.

The thrust bearing (set 3) on the ball nut were specially designed for this particular application to meet the available envelope and the bearing manufacturer used the above criteria for their design and should therefore be capable of meeting the 72-hour endurance life.

The motor bearings are somewhat overloaded on the basis of the above design criteria. These bearings have performed well during tests but on redesign should probably be increased in rating unless future endurance testing indicates them to be satisfactory.

P/N	Use	Bearing (1) Load	RPM (2) Average	500 Hr. Min. Capacity @ Ave. RPM
2775583	Motor Rotor	330	3060	218 Lbs.
2775581	Planet Bearings	170	1900	170
2775582	Cage Bearing	439	680	1000
2775580	Idler Bearing	690	680	700
2775568	Thrust Bearing	23, 200	218	23, 200

(1) Load at Motor Rated Torque 147 in. lbs.

(2) Average Endurance RPM 1. 45 in. /sec.

Table 3-2. Bearing Loads and Ratings

3.8 ACTUATOR STRUCTURAL STIFFNESS

In order to provide an actuator with the minimum effect on total system spring rate, it was decided to use a target criteria of 6×10^6 lbs./in. for the structural spring rate. Considering an overall length of twenty-three inches and steel with a modulus of elasticity of 29×10^6 this would mean an average cross section of:

$$\frac{\text{Rate}\left(\frac{\text{lb.}}{\text{in.}}\right) \times \text{Length (in.)}}{\text{E (psi)}} = \text{Cross Section In.}^2$$

$$\frac{6 \times 10^6 \times 23 \text{ in.}}{29 \times 10^6} = 4.75 \text{ in.}^2$$

Considerable compromise was necessary, however, in certain areas such as the rod end, gearing, etc., but wherever possible this cross sectional area was maintained. See Section 4.5 for stiffness test results.

3.9 RECIPROCATING BEARING

The specification requires operation with a side load of 5,000 pounds applied to the center of gravity of the actuator. Assuming the center of gravity to be halfway between the reciprocating bearing and the head end of the actuator, the reciprocating bearing must be capable of withstanding a load of 2,500 pounds. Rulon LD was selected as the bearing material to be used. The maximum velocity of the bearing is 3.3 in./sec. or 16.5 ft./min. The projected diameter of the bearing is 3.735 inches and the material is capable of 20,000 PV factor. The width of bearing material was calculated as follows:

$$PV = \frac{\text{Load (Pounds)}}{\text{W (in.)} \times \text{Dia. (in.)}} \times \text{Velocity (ft./min.)}$$

$$20{,}000 \text{ psi ft./min.} = \frac{2500 \text{ pounds}}{W \times 3.735} \times 16.5$$

$$W = \frac{2500 \times 16.5}{40{,}000 \times 3.735}$$

$$W = 0.554$$

A width of .560 was used.

3.10 DYNAMIC SEAL

The dynamic seal requirement was recognized as one of the most difficult considering the -250°F temperature. Contact was made with ten different seal companies and only three offered designs. Of these designs, the Bal Seal was chosen because of its small cross section and it appeared to be the best seal with respect to sealing capability. The prime consideration was seal life. A component development test was initiated early in the actuator development program since this appeared to be a significant problem.

The component development on the dynamic seal is covered in Section 4. 4 of this report. In short, it was found that the outer lip of the seal would collapse the garter spring expander at approximately -170°F. A new expander was designed which consisted of a solid split ring which was spring loaded axially to keep the outer lip in contact with the bore.

This new design would seal to -250°F but it was not applied to the actuators since the design had progressed past the point where the new design could be installed within available space.

3. 11 SNUBBER DESIGN

The snubber must absorb all the energy developed at the actuator in the scram mode. The energy to be absorbed is comprised of the load and actuator kinetic energy and the potential energy of the motor. Part of this energy will be dissipated within the actuator and test stand or vehicle structure and it is apparent then that the snubber spring will actually not deflect the full 0. 1 inch available. It is anticipated that the actuator testing will provide the data showing the proportions of the energy absorbed by structure and the snubber and that the required snubber spring rate will be reduced accordingly.

The spring material is AMS 5528 or AMS 5644B (17-7 PH plate or forging). This material was selected for its availability, high stress capability and retention of ductility at cryogenic temperatures. The maximum spring design stress was set at 300, 000 psi.

3. 11. 1 Design Calculations

T - Kinetic Energy

T_L - Kinetic Energy of the Load - In. Lb.

T_A - Kinetic Energy of the Actuator - In. Lb.

m - Mass - lb. sec. ^{2}/in.

V_M - Potential Energy of Motor at the Snubber - In. Lb.

Parameters:

Maximum Velocity	3. 3 in. /sec.
Load Mass	116. 67 lb. sec.^{2} /in.

Actuator Mass	150. 36 lb. sec.^{2}/in.
Minimum Actuator Force	42, 000 lbs.
Load Friction	0
Actuator Mechanical Efficiency	80%

Energy Input to Snubber:

Kinetic Energy (T) $= 1/2\ mv^2$

Kinetic Energy of Load (T_L) $= \dfrac{1400 \times 3.3^2}{2 \times 12} = 635.3$ in. lb.

Kinetic Energy of Actuator (T_A) $= \dfrac{150.36 \times 3.3^2}{2} = 843$ in. lb.

Potential Energy of Motor (V_M):

V_M = Force x Distance

Assume torque-speed curve of Motor is Linear then:

$$V_M \cong \frac{42,000}{2} \times 0.1 = 2100 \text{ in. lb.}$$

Total Energy to be Absorbed:

$$= V_M + T_A + T_L = 3578.3 \text{ in. lb.}$$

Required Spring Rate:

The work done in compressing a spring $= 1/2\ K\ F^2$

where K = Spring Rate - lb. /in.

F = Spring Deflection - inches

Equating work done by spring to energy to be absorbed

$$1/2\ K\ F^2 = 3578.3$$

let $F = 0.1$

then $K = \dfrac{3578.3 \times 2}{0.01} = 705,660$ lb. /in.

Spring Force (P):

To facilitate calculation of spring dimensions and the spring force at F = 0.1

$$P = FK = 0.1 \times 705,660 = 70,566 \text{ lbs.}$$

Spring Package Dimensions:

The space available within the actuator dictated the inner and outer spring diameters. To realize the spring force and deflection within the permissible stress range it was found necessary to use three springs in series.

To use one spring assembly for snubbing both directions of actuator travel, a fourth spring is required to maintain the spring assembly contact surfaces in the necessary positions. However only three springs are active in either direction. Figure 3-8 shows this arrangement.

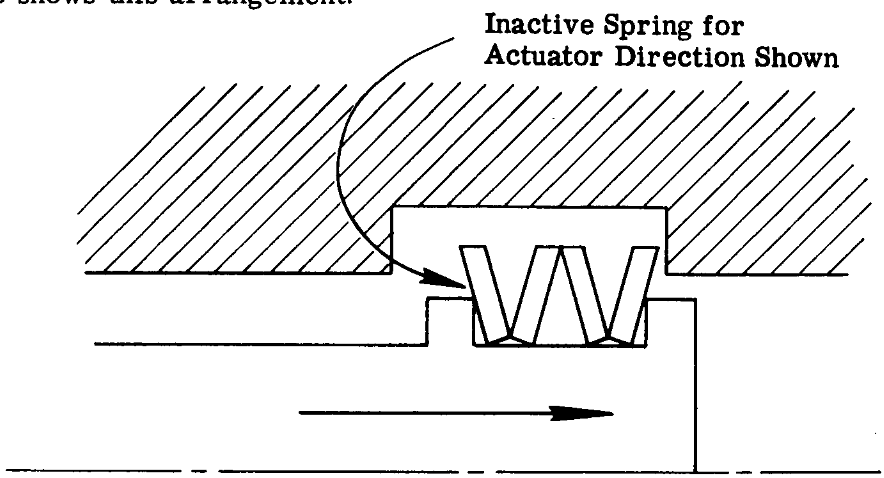


Figure 3-8. Schematic Diagram of Snubber

Spring Calculations:

Outside Diameter			5.00 inches
Inside Diameter			3.750 inches
P	-	Load - lbs.	
E	-	Modulus of Elasticity	30×10^6 psi
p	-	Poisson's Ratio	.29
h	-	Inside Height - inches	0.050

F - Deflection - inches 0.033

E - Thickness - inches

a - One-Half Outside Diameter in Inches

S - Stress - psi

$\left.\begin{array}{l} C_1 \\ C_2 \\ M \end{array}\right\}$ Logarithmic equations, values taken from Associated Spring Corp. chart

Q - Empirical Form Factor = .95

The Basic Formula (Associated Spring Corporation)

$$P = K_1 \left[(h-F)(h-F/2) E + E^3\right]$$

$$S = K_1 \left[C_1 (h-F/2) + C_2 E\right]$$

where $K_1 = \dfrac{E F}{(1-p^2) Ma^2}$

The spring shape used differed from that used for the standard formula as follows. The outer spring diameter was machined to cylindrical form and the contact edges were machined to form a ring contact 0.030 to 0.060 wide. A factor Q was obtained experimentally, and may only be considered valid for a spring of the proportions used.

Since three springs are used in series, each spring must deflect 0.033 inch for a 70,566 pound load. These springs will be operating unlubricated.

Find K_1

$$K_1 = \frac{30 \times 10^6 \times 0.033}{0.916 \times 0.4 \times 6.25} = 4.33 \times 10^5$$

Find E as a function of P

Try E = 0.532

$$P = K_1 \left[(h-F)(h-F/2) E + E^3\right] Q$$

$$P = 4.33 \times 10^5 (5.69 \times 10^{-3} \times 0.532 + 0.532^3) 0.95$$

$$P = 63,200 \text{ pounds}$$

Find Stress:

$$S = K_1 \left[C_1 (h-F/2) + C_2 E\right]$$

$$S = 4.33 \times 10^5 \left[1.05 (0.050 - 0.0165) + 1.1 \times .532\right]$$

$$S = 268,500 \text{ psi}$$

SECTION IV

EVALUATION OF ACTUATOR AND ITS COMPONENTS

4.1 DEVELOPMENT HISTORY

The Bendix Corporation has for some years been involved in developing electro-pneumatic actuators for use in extreme environments. Experience has been gained with torque motors, servo valves, pneumatic motors, bearings, gears, etc. The design and development of the J-2 actuator made use of this extensive backlog of development. The past developments used and the additional development effort required is noted for the various components as follows:

4.1.1 Torque Motor and Servo Valve

Bendix has designed and developed torque motors and also evolved torque motor designs with many vendors. The low power requirement for this torque motor (250 mw) was the main problem to overcome.

Several servo valve configurations have been developed for electropneumatic servo systems. It was necessary to select a configuration best suited to this application and size the valve to provide the necessary capacity. The greatest problem, however, was in combining the torque motor and servo valve characteristics to obtain a valve that would be stable with air, nitrogen, and hydrogen over the wide range of temperatures.

4.1.2 Servo Motor

Pneumatic servo motors of gear and vane types have been developed by Bendix to operate on various gases from cryogenic temperatures to 1200° F. In this case, a double lobe vane motor was selected for its low inertia and low weight, realizing that friction would be a problem with respect to the system deadband. A breadboard motor was designed, built, and evaluated to aid in sizing the motor for this application.

4.1.3 Ball Screw

Bendix has conducted development programs on ball screws for both cryogenic and high temperature applications. With respect to ball screws for cryogenic temperatures, previous test programs have indicated that the normal well lubricated load rating should be reduced to 25% for use with known methods of dry-film lubrication. This much derating would require a larger screw than could be tolerated in the J-2 actuator. The screw selected was derated to 36% and a development program was initiated at Bendix Products Aerospace Division to provide a better means of lubrication.

4.1.4 Dynamic Seal

Little background knowledge was available with respect to reciprocating dynamic seals. Contacts with many seal manufacturers indicated that there was only one off-the-shelf seal available for this type of application. A development program was initiated to test the "Bal" seal for this application. Included in this program was side load testing of the reciprocating bearing.

4.1.5 Ball Bearings

The Bendix Corporation has gained considerable knowledge with respect to the use of ball bearings at temperature extremes beyond normal lubricant capabilities. The cryogenic temperature requirement of this actuator required the use of the Bendix-developed self-lubricating ball retainer design. The ball retainer is a solid type made of Du Pont's SP-1 Super Polymer. Ball action against the retainer transfers minute quantities of SP-1 to the races to provide a dry-lubrication film. Development tests indicate this material to be far superior to any other method of dry-film lubrication tested with respect to life and required bearing derating.

4.1.6 Gear Lubrication

Previous development programs have indicated that the best dry-film lubrication for gear teeth for this application is General Magnaplate Corporation's Hi-T-Lube. The material is a proprietary film composed of gold, silver, and MoS_2 0.0003 to 0.0007 inch thick and, in this case, is applied to alternate gears in the system.

4.2 SERVO COMPONENT DEVELOPMENT PROGRAMS

4.2.1 DC Servo Amplifier

4.2.1.1 General Description and Specifications

4.2.1.1.1 General Description. The Model 9513 is a special purpose DC Amplifier assembly designed to sum three separate ground referenced input signals and drive an external two-wire permanent magnet torque motor. The assembly consists of a wired rack adapter containing a Model 9552 Voltage Amplifier, a Model 9553 Feedback Board and a Model 9554 Power Booster. Operating power is obtained from external DC power supplies.

4.2.1.1.2 Specifications. Typical performance at 25°C.

Gain: .014 to .070 amps/volt, adjustable

Gain Stability: ±10% of setting

Input: Three single ended summing inputs

Input Impedance: 200K ohms, each input

Maximum Input: ±30 volts, each input

Frequency Response: Figure 4-1

Rated Output: ±55 volts peak at 50ma peak

Voltage Limit: ±60 volts nominal

Current Limit: ±55ma nominal

Output Offset: Adjustable to zero

Output Drift: ±.05 ma/hr at constant temperature after warmup

Ripple & Noise: ±.01ma peak to peak

Static Linearity: ±5%

Temperature Range:
Operating: +15°C to +35°C
Storage: 0°C to +50°C

Supply Required:
Voltage: +70VDC @100ma peak
-70VDC @100ma peak
Accuracy: ±1%
Regulation: ±0.1%
Ripple & Noise: 250μv rms
Impedance: 0.1 ohms

Torque Motor: Midwestern Instruments, Inc. Model 39-4

Package:
Type: Half Rack
Size: 3-1/2" H x 9-1/2" W x 10" D

Connector:
Input: AN3102A-14S-5P
Output: AN3102A-14S-5S
Power Supply: AN3102A-14S-2P

Controls: Gain
Zero

4.2.1.2 Operating Instructions

4.2.1.2.1 Installation. The Model 9513 is designed for relay rack mounting and is constructed as a half-rack width assembly. The Model 9552 Voltage

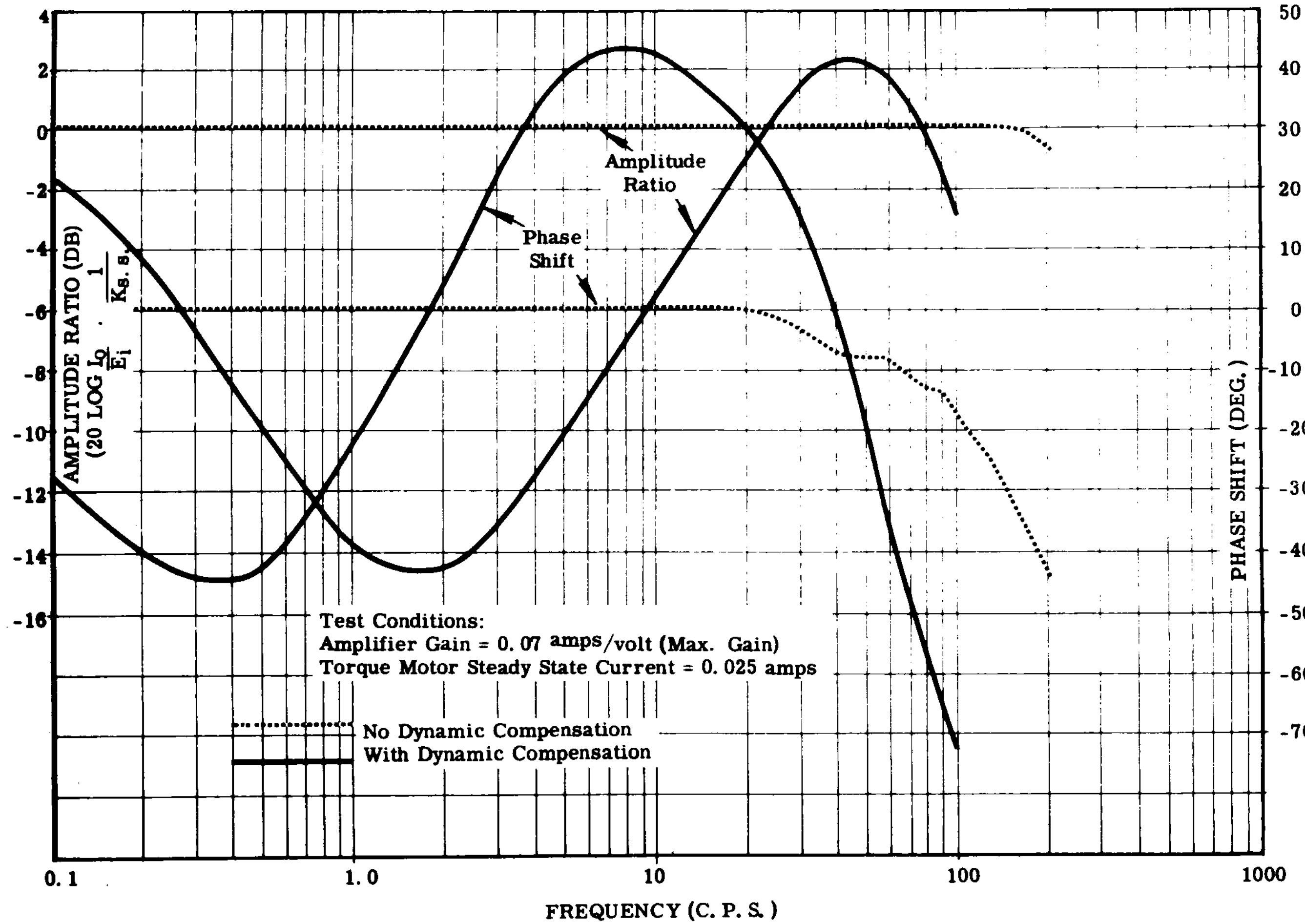


Figure 4-1. Servo Amplifier Frequency Response.

Amplifier, Model 9553 Feedback Board and Model 9554 Power Booster are packaged as Burr-Brown 1600 Series Plug-In-Modules. Each of these is secured to the rack adapter frame by means of two knurled captive panel screws. When the Plug-In-Modules are installed in the rack adapter, it is important that they be located in the correct position. Damaging power supply potentials may be applied to these units if they are improperly installed.

4.2.1.2.2 External Connections. External connections from the signal source, the torque motor coils, and the power supplies are made to pins on the mating connectors as indicated below:

J6	Input Connector	AN3106A-14S-5S
	Input 1	Pin E
	Input 2	Pin D
	Input 3	Pin C
	Signal Common	Pin B
	Chassis Gnd.	Pin A
J4	Output Connector	AN3106A-14S-5P
	Output to Coil	Pin D
	Coil Return	Pin E
	Current Monitor	Pin C
	Signal Common	Pin B
	Chassis Gnd.	Pin A
J5	Power Supply Connector	AN3106A-14S-2S
	Positive Supply, +70VDC	Pin D
	Negative Supply, -70VDC	Pin C
	Signal Common	Pin B
	Chassis Gnd.	Pin A

The following wiring precautions should be observed when external connections are made to the source and the load. The input leads should be shielded to minimize pickup of spurious noise signals. The signal common return connection should be made to a single point in the external circuit to eliminate possible ground loops. Signal common should be electrically connected to chassis ground either at the Model 9513 or at some point in the external circuit.

4.2.1.2.3 Power Supply Requirements. The Model 9513 obtains its operating power from a pair of power supplies that deliver +70VDC and -70VDC. The two supplies are connected in series with their junction or center tap electrically connected to signal common, Pin B, on the power supply connector.

4.2.1.2.4 Phase Compensation Modification. The partially connected phase compensation may be utilized to alter the closed loop amplifier gain characteristics when the center legs of the four tee networks are connected to common. To perform this modification, proceed as follows:

Gain access to the interconnecting wiring between J_4, J_5, J_6, and the 1600 Series Module Sockets by pulling the rear panel of the 9513 rack adapter clear of the chassis. Connect wire jumpers from pins B, C, D, and E on the socket for the Model 9553 Feedback Board to signal common. Replace the rear cover.

The amplifier will now have a gain characteristic that is flat from DC to .15cps, falls at 6db/octave from .15cps to 1.5cps flat from 1.5cps to 2.0cps, then rises 6db/octave from 2.0cps to 20cps and remains flat from 20cps to 80cps approximately. Should it be desirable to modify these characteristics, refer to manufacturer's instruction manual Model 91513, DC Servo Amplifier for compensation component location.

4.2.1.2.5 Manufacturer. Burr-Brown Research Corporation, Box 6444, Tucson 16, Arizona.

4.2.1.3 DC Servo Amplifier Dual Power Supply

4.2.1.3.1 General Description. The Model 630 dual DC power supply is designed to supply two separate regulated DC voltages.

4.2.1.3.2 Specifications

Temperature: Continuous duty at full load from 0° to 55°C ambient

Output Voltage Each Unit: 70VDC adjustable ± 1 volt

Rated Current Each Unit: 125ma

Ripple: Less than 0.002 volts rms

Regulation: Load 0.1%, Line 0.05%

Impedance: 0.08 ohms @1 Kc

Supply Voltage: 105-125V, 50 to 400cps single phase

Connector: 11 pin octal plug

External Connections:
- AC Output: Pins 1 % 2
- Output No. 1: Pins 10(+) 9(-)
- Output No. 2: Pins 3(+) 4(-)

4.2.1.3.3 Polarity. Outputs are floating. Individual section may be connected as a separated power supply. Either negative or positive side of one section may be grounded irrespective of the other section.

4.2.1.3.4 Manufacturer. Acopian Technical Co., 927 Spruce Street, Easton, Pennsylvania.

4.2.1.4 Amplifier And Power Supply Installation. The amplifier and power supply are mounted and interconnected on a common chassis. The chassis is sized for a standard 19" relay rack.

4.2.1.5 Amplifier Dynamic Configuration. The amplifiers, as shipped, have the compensation circuits.

4.2.2 Torque Motor

A comparative evaluation of three torque motors was conducted early in the program. These motors were: Midwestern Instruments Model 39-4, Servotronics Model 29-2-32, and a Bendix-developed torque motor modified to meet the requirements of the J-2 program. Figures 4-3 to 4-10 have been selected as typical performance characteristics of these torque motors.

Figure 4-2 shows the spring rates of the various torque motors. The Midwestern torque motor is stiffest with a rate of 52.1 lb/in. while the modified Bendix motor was softest with a rate of 19.7 lb/in. Normally a high spring rate torque motor is desirable, since this will tend to reduce valve instability.

Performance characteristics of the Servotronics motor, at room temperature, is superior to the Midwestern motor. However, it was found that the lower spring rate of the Servotronics torque motor made it more difficult to obtain a stable servo valve and Midwestern torque motors were shipped on both actuators.

4.2.3 Servo Valve

4.2.3.1 Introduction

Figures 4-14 to 4-23 have been selected as typical performance characteristics which can be expected from the servo valve. Figure 4-11 shows the servo valve assembly as it was tested. The tubular assemblies bolted to the ends of the valve body are end caps specially designed to facilitate valve evaluation. Figure 4-12 is an end view of an experimental end cap showing the location of the spool proximity transducer and the removable damping chamber orifice. A pressure transducer, or gage, is normally connected to the capped fitting. The volume of the end cap can be varied by adjusting and locking the large screw projecting from the end of the cap. The screw positions a sealed piston located in the tubular body. Figure 4-13 shows the disassembled experimental valve.

4.2.3.2 Power Stage

Valve performance was evaluated by placing load orifices in the valve motor ports, applying a command signal, and observing motor port pressure fluctuations. Proper orifice size selection enables a good approximation of servo valve-vane motor performance. Additional valuable valve data is obtained if motor port pressure data is recorded with the motor ports blocked and wide open. This information is shown in Figures 4-14 to 4-17 for an 800 psig nitrogen supply.

(Text continued on page 4-22)

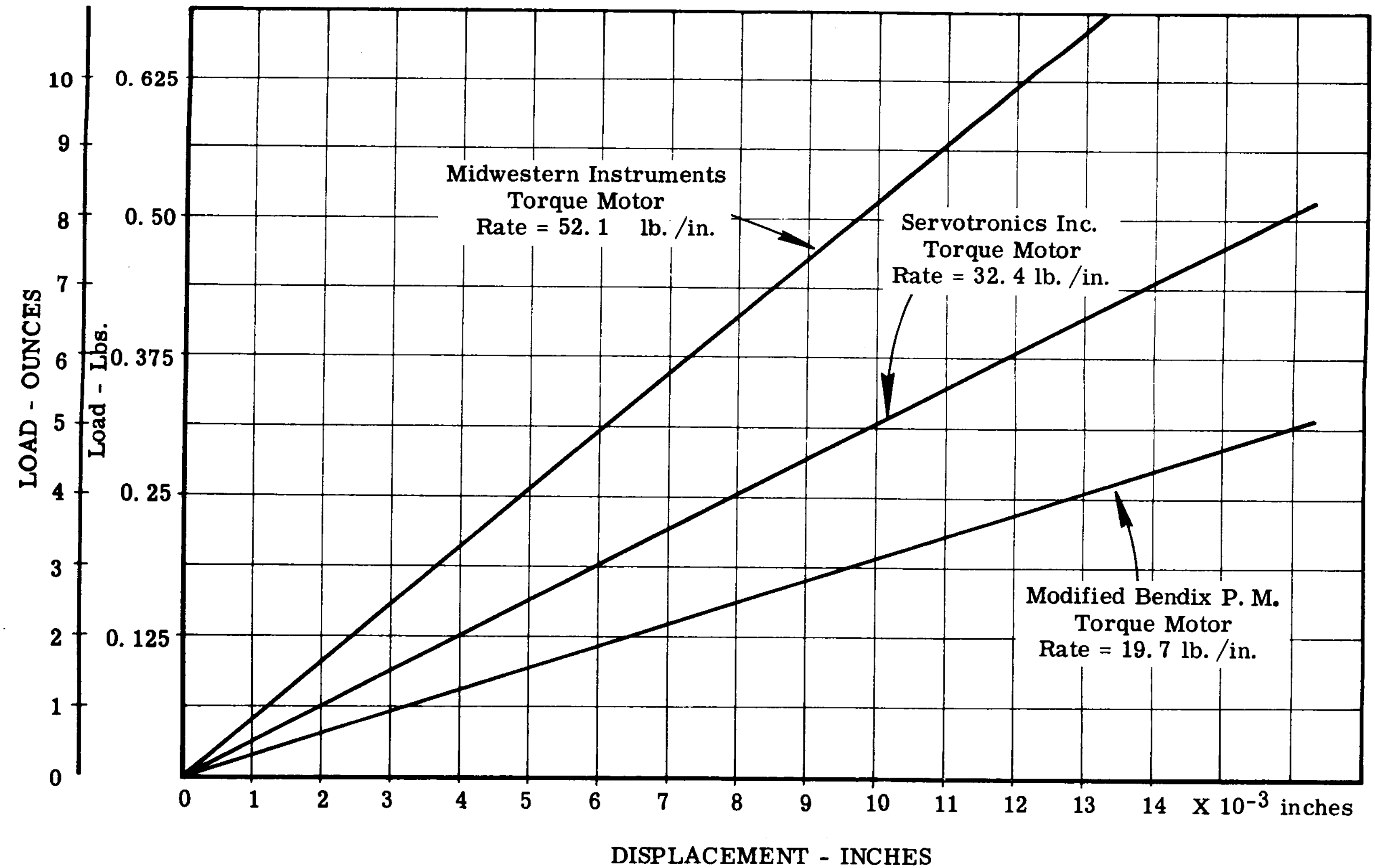


Figure 4-2. Torque Motor Spring Rates.

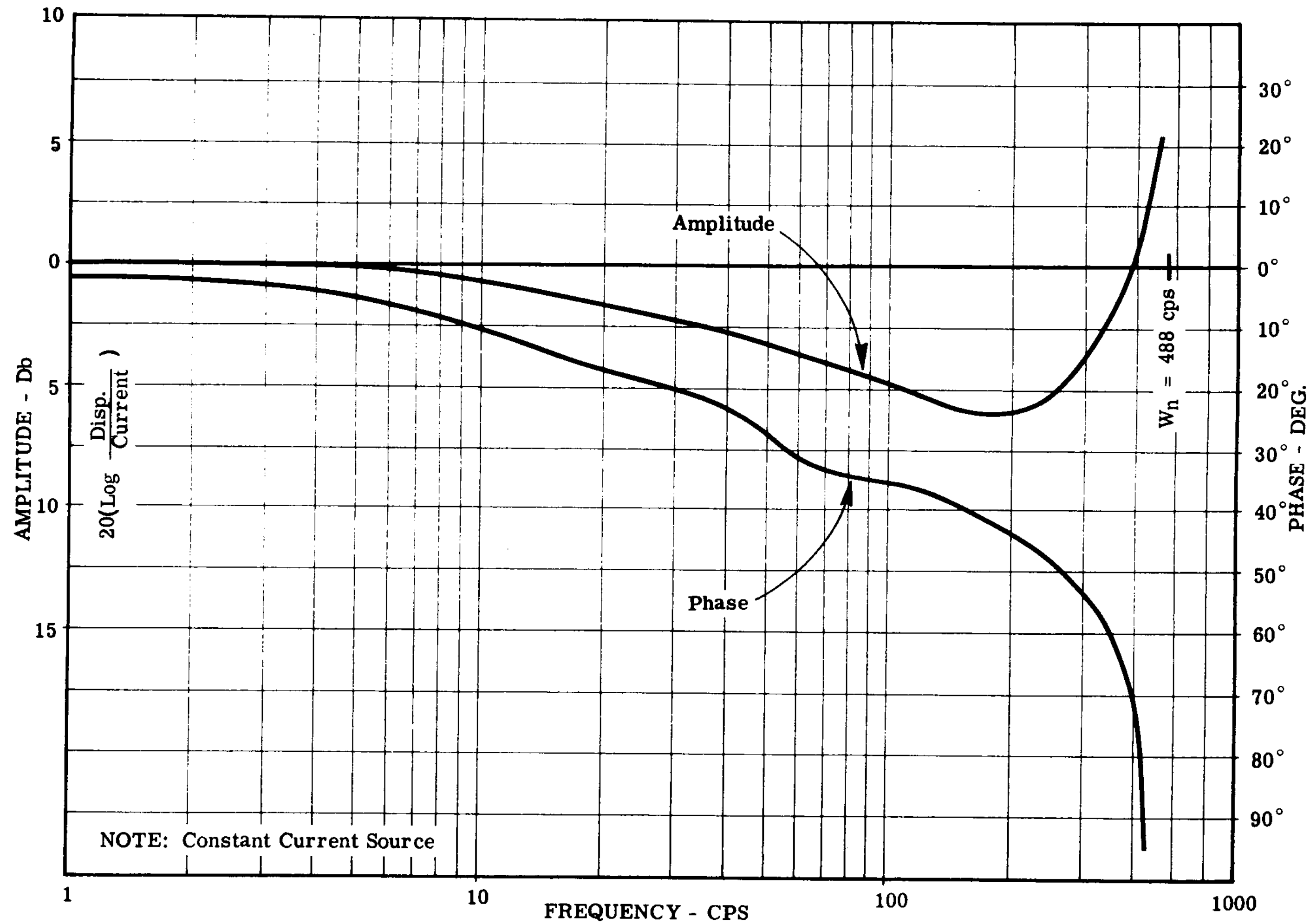


Figure 4-3. Frequency Response - Midwestern Instruments Torque Motor.

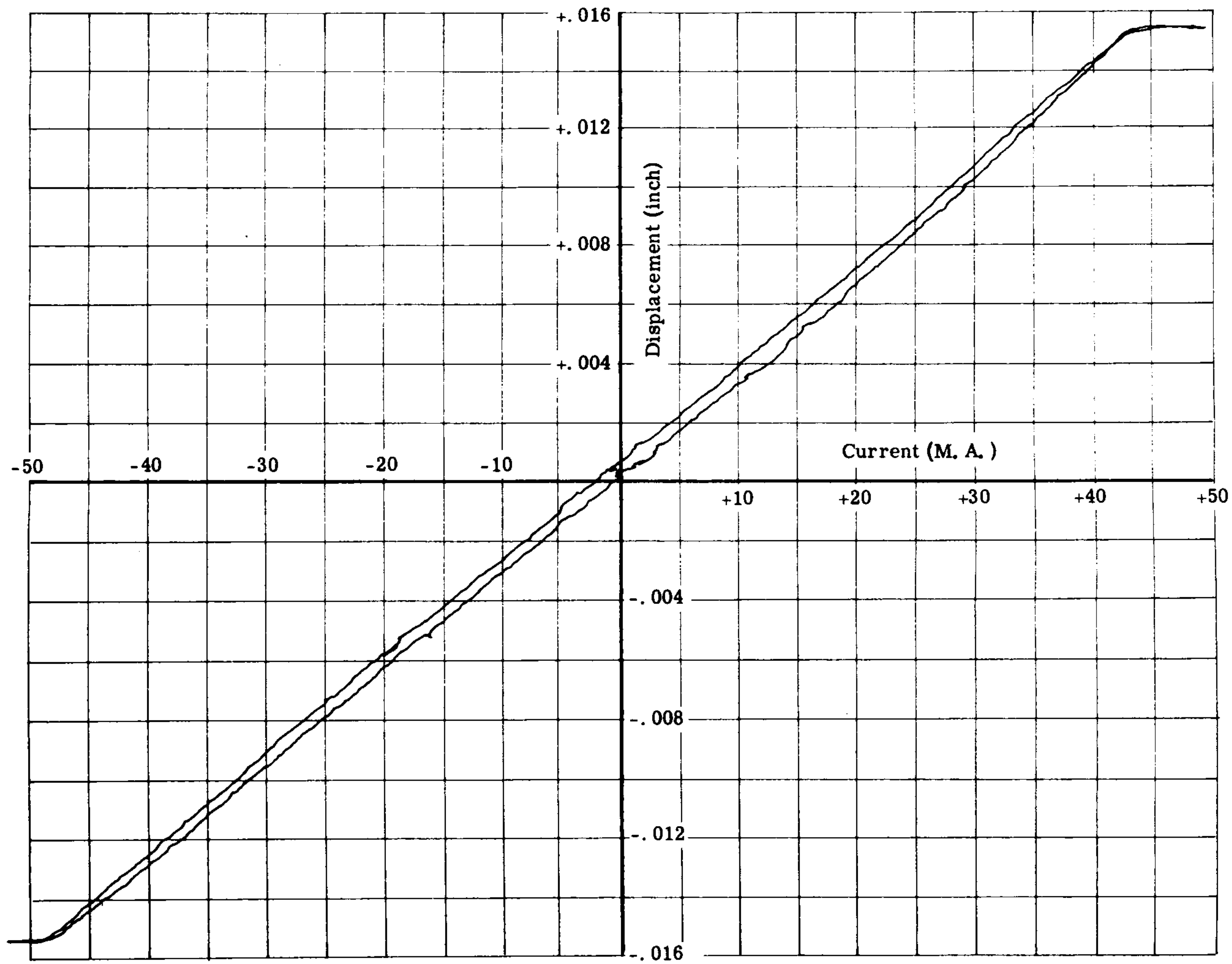


Figure 4-4. Gain Curve - Midwestern Instruments Torque Motor.

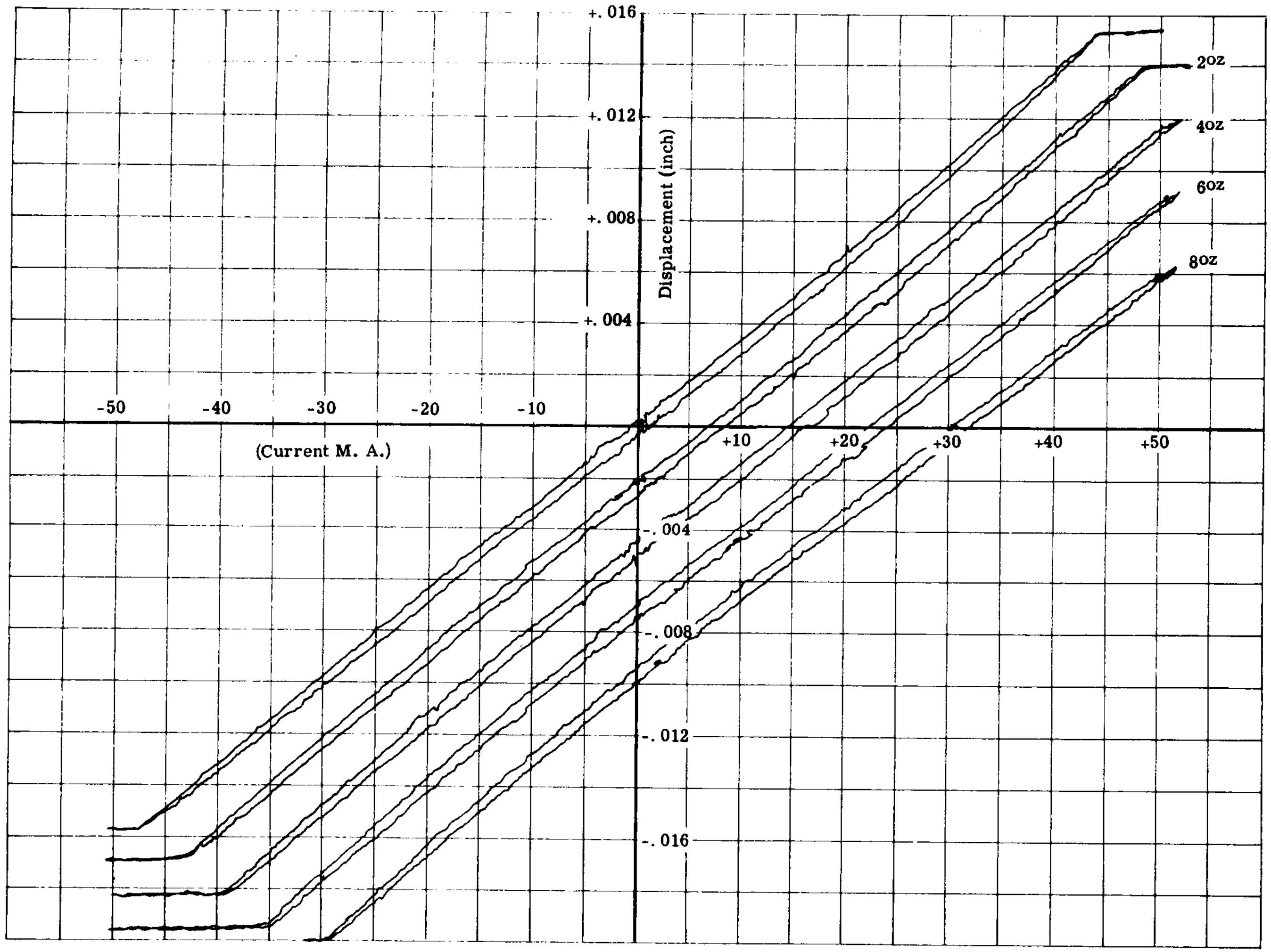


Figure 4-5. Force-Displacement Curve - Midwestern Instruments Torque Motor.

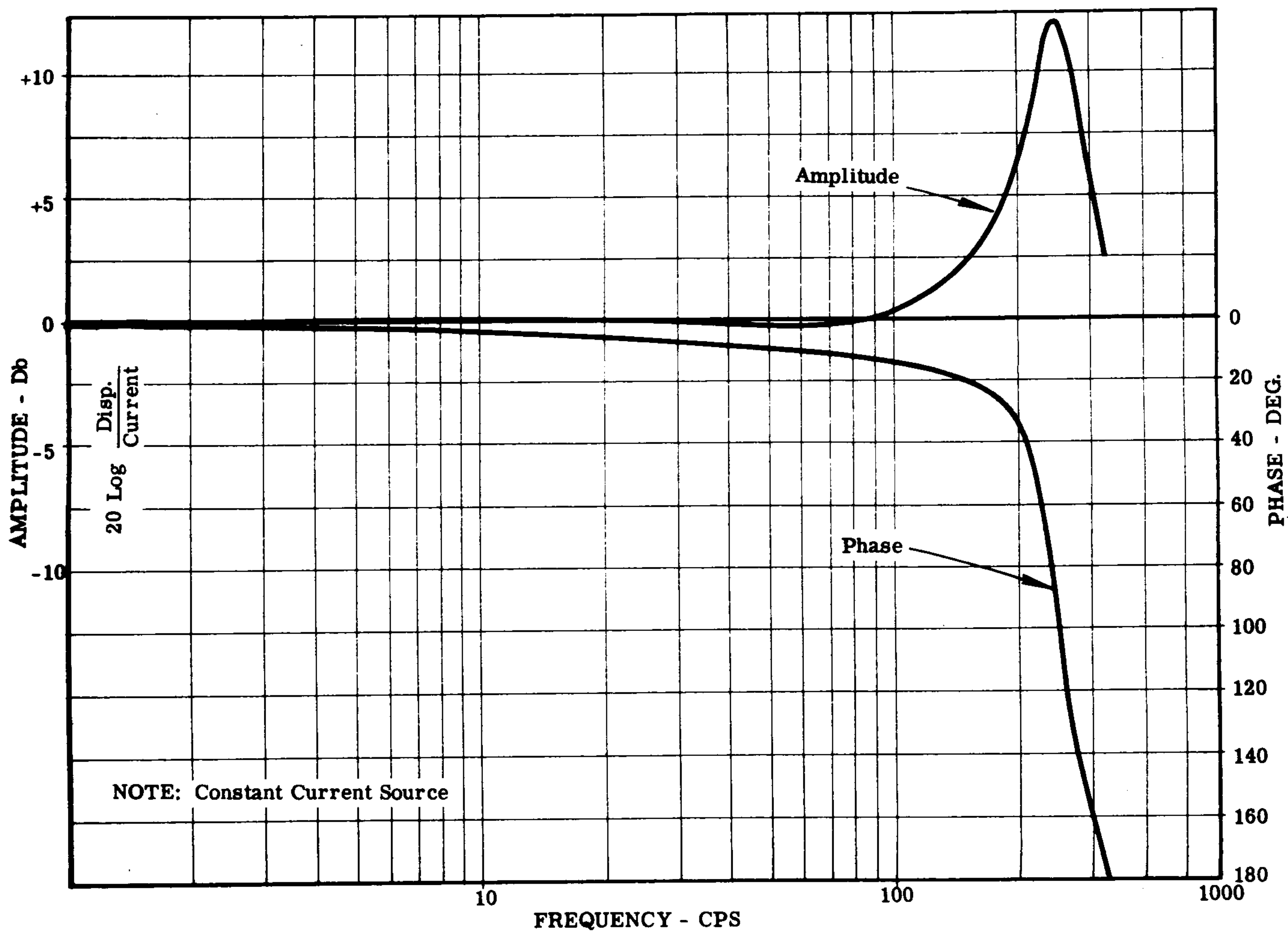


Figure 4-6. Frequency Response - Servotronics Torque Motor.

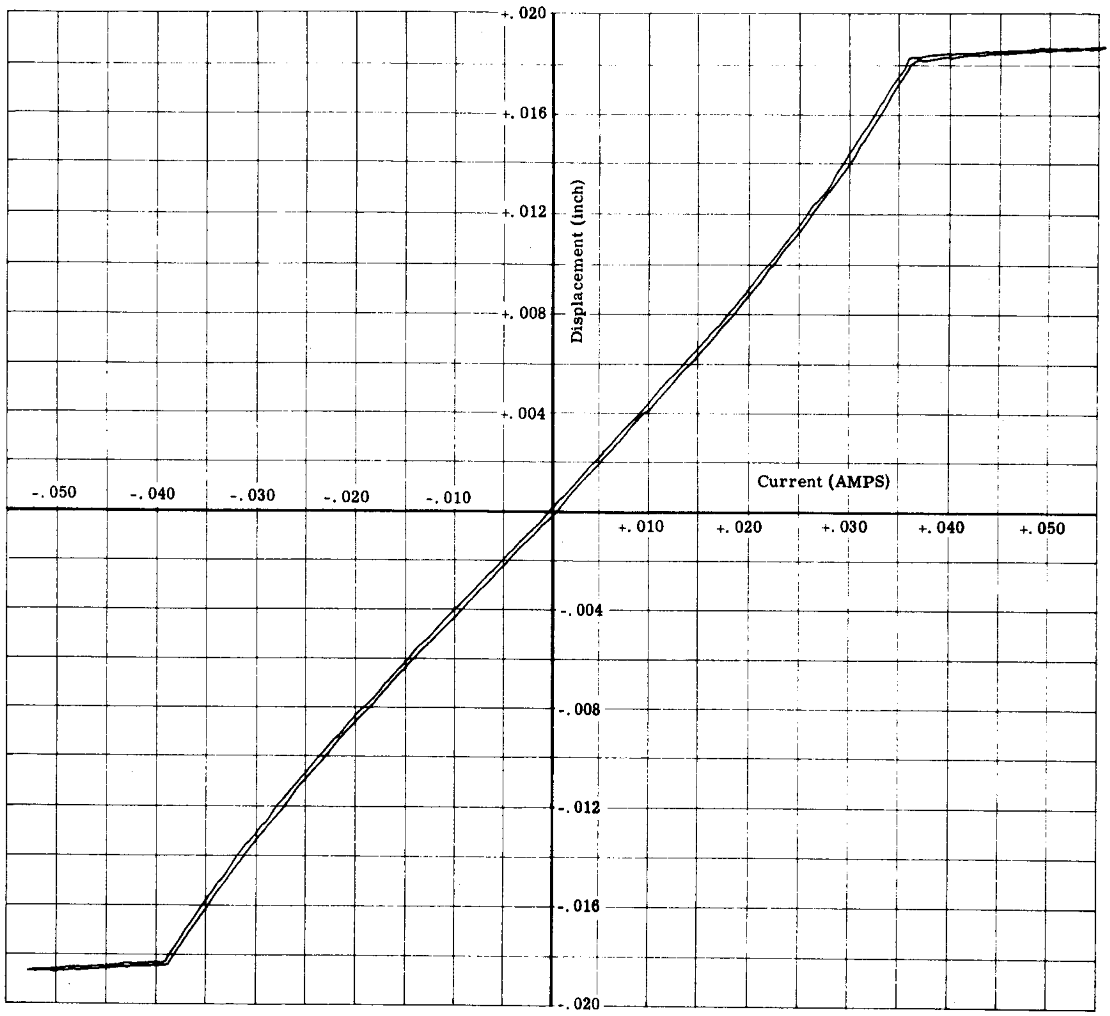


Figure 4-7. Gain Curve - Servotronics Torque Motor.

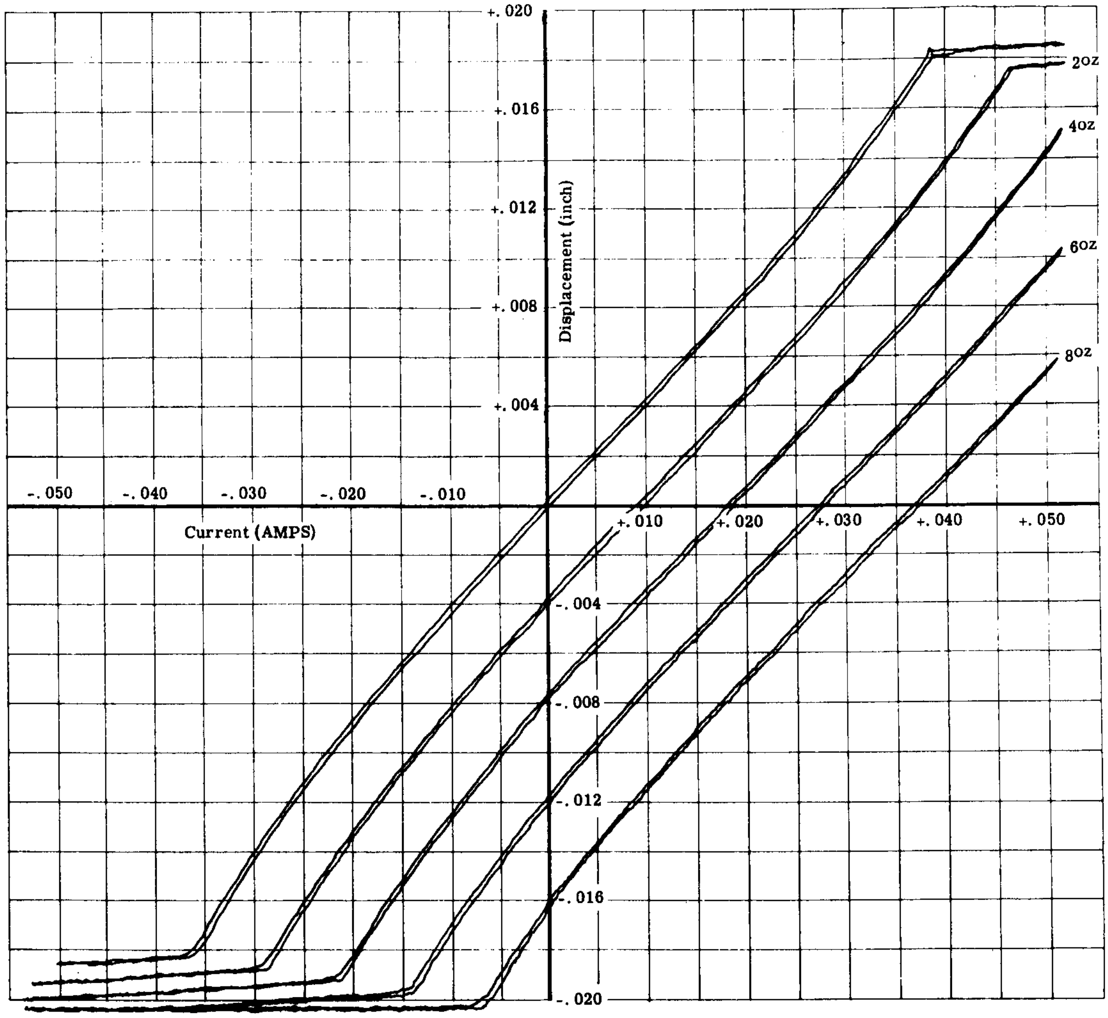


Figure 4-8. Force - Displacement Curve - Servotronics Torque Motor.

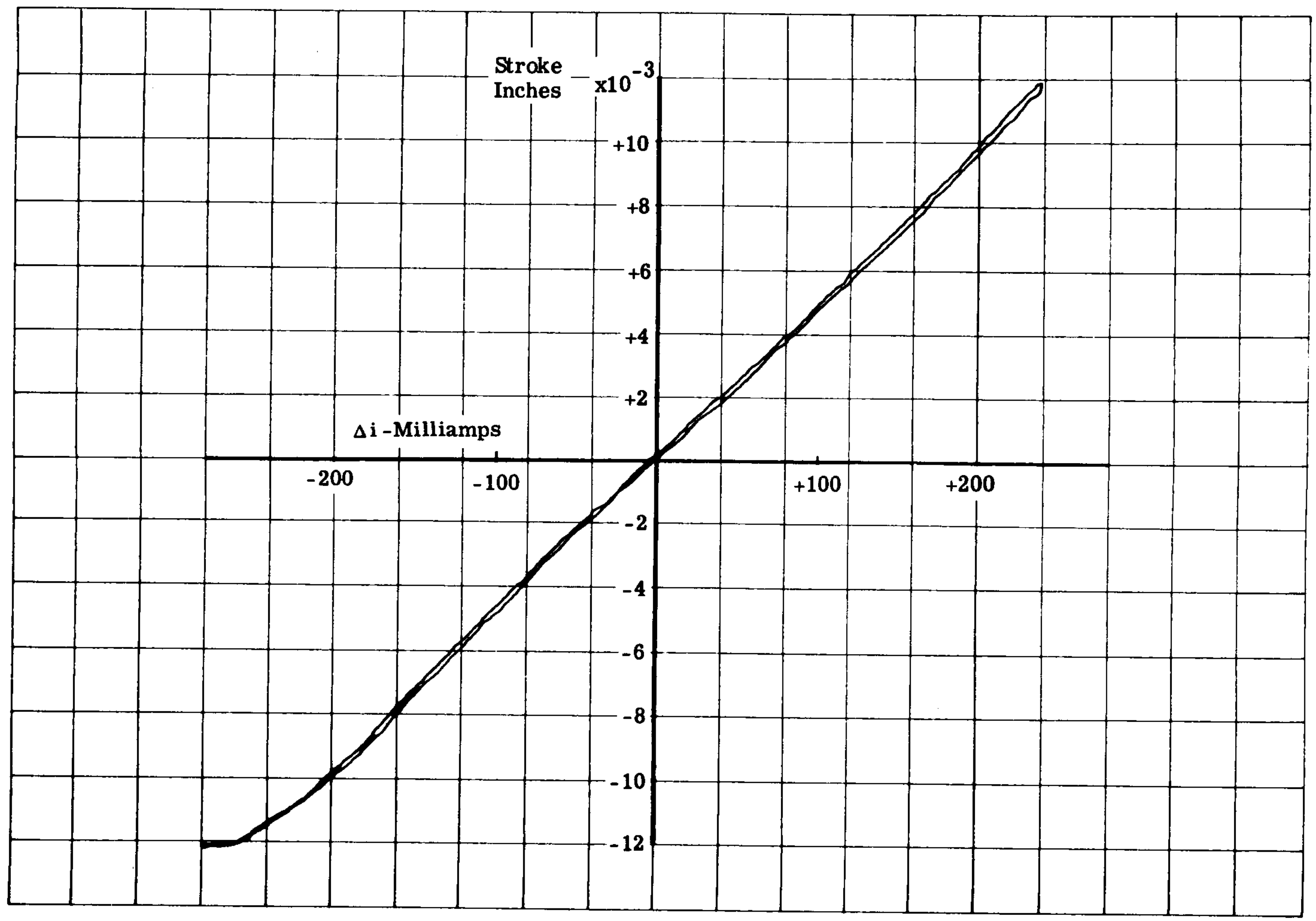


Figure 4-9. Gain Curve - Modified Bendix P.M. Torque Motor.

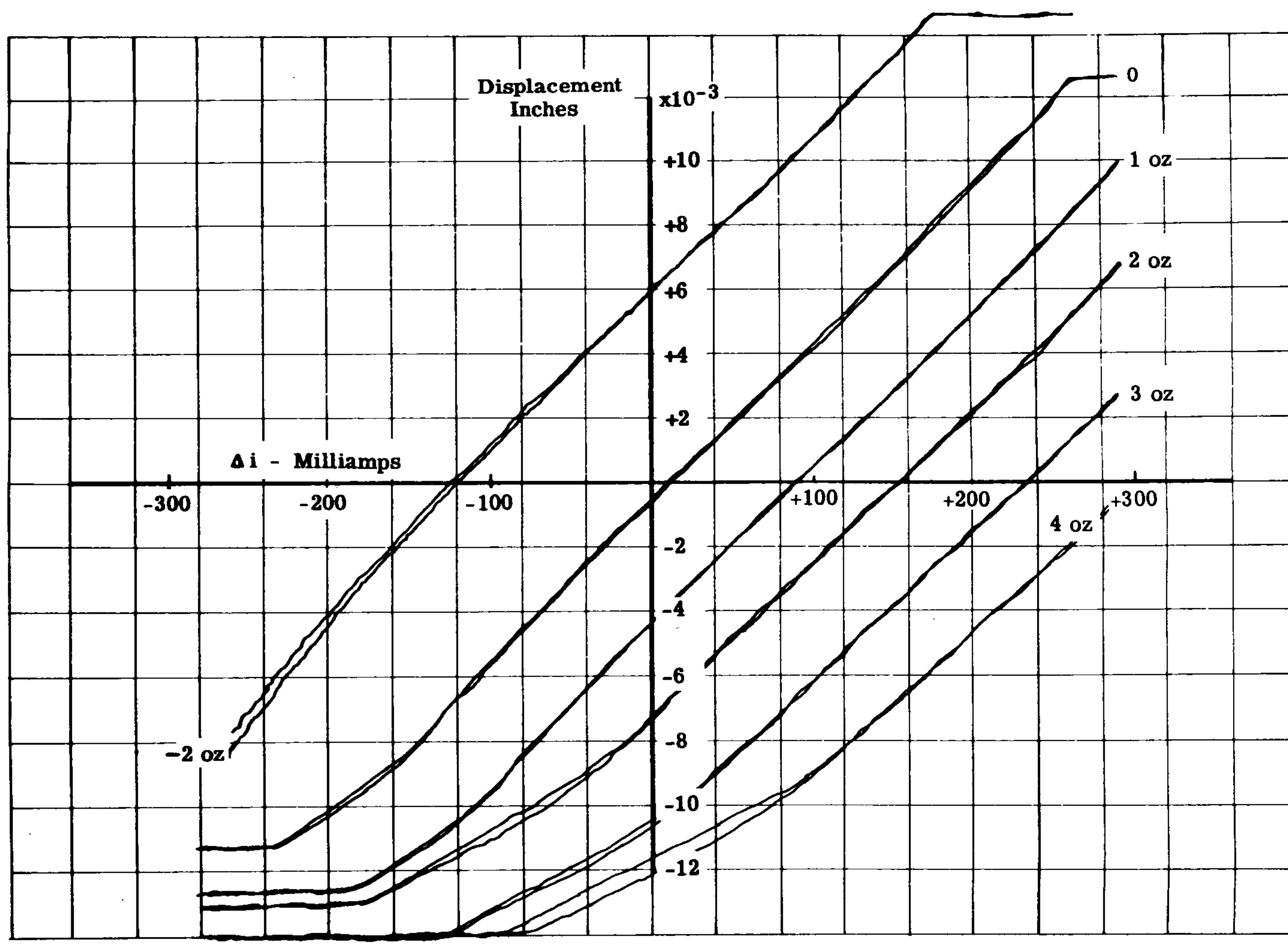


Figure 4-10. Force-Displacement Curve - Modified Bendix P.M. Torque Motor.

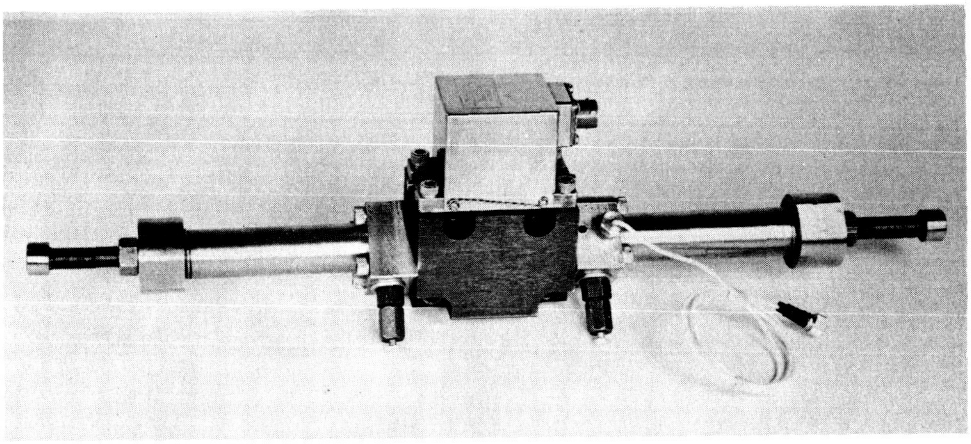


Figure 4-11. Servo Valve Assembly As Tested.

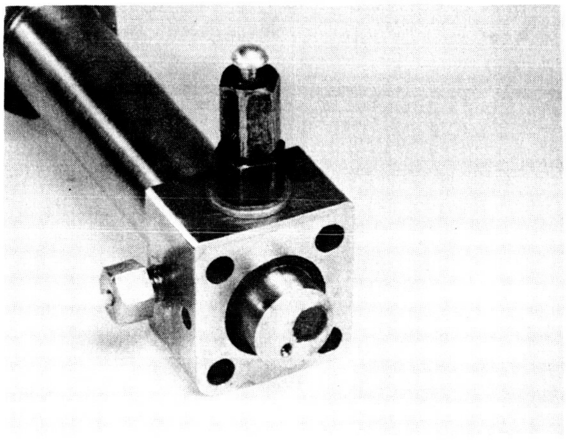


Figure 4-12. End View Of Experimental End Cap.

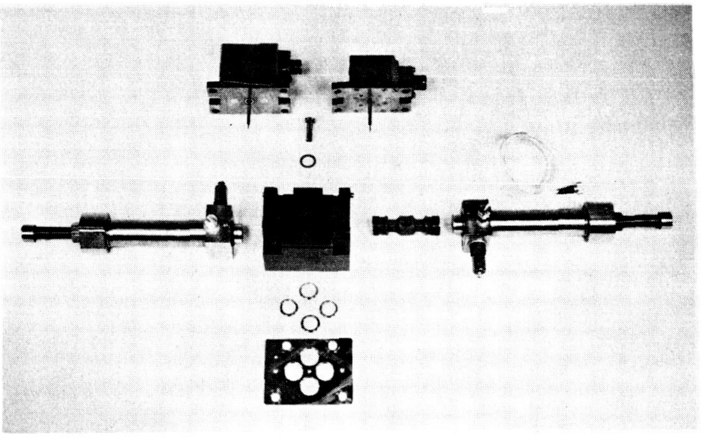


Figure 4-13. Exploded View Of Experimental Servo Valve.

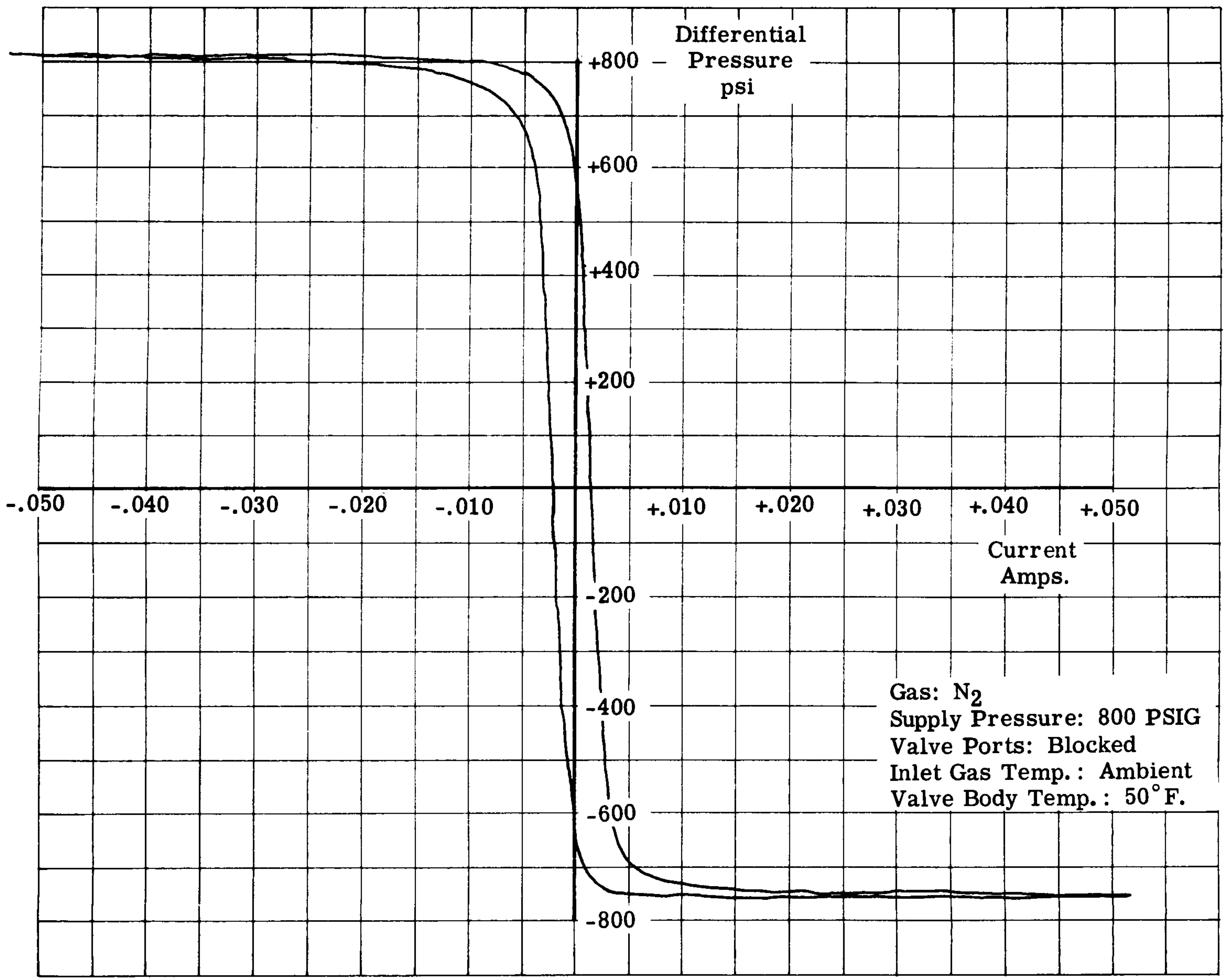


Figure 4-14. Motor Port Pressure Sensitivity (Blocked Ports).

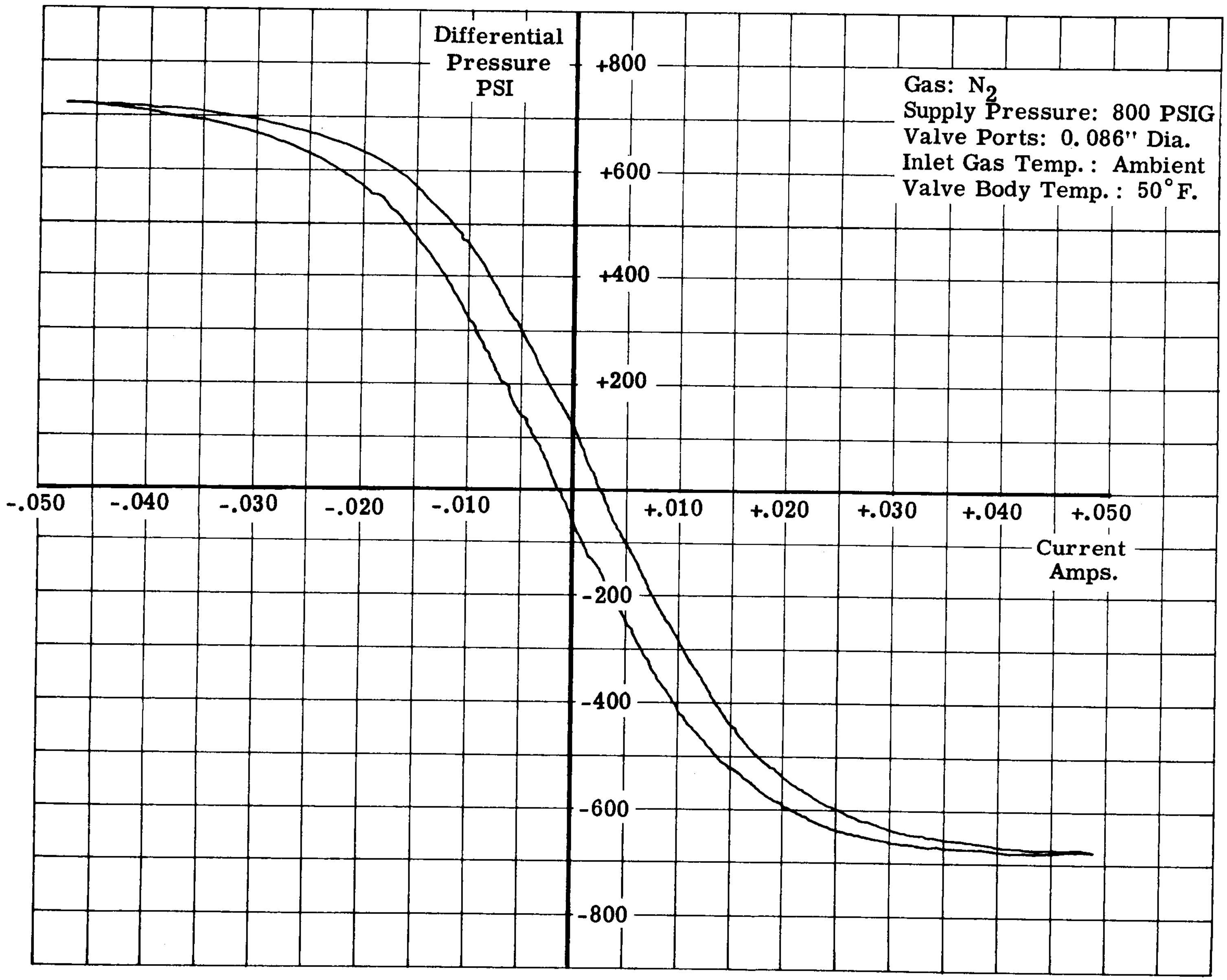


Figure 4-15. Motor Port Pressure Sensitivity (0.086" Diameter Ports).

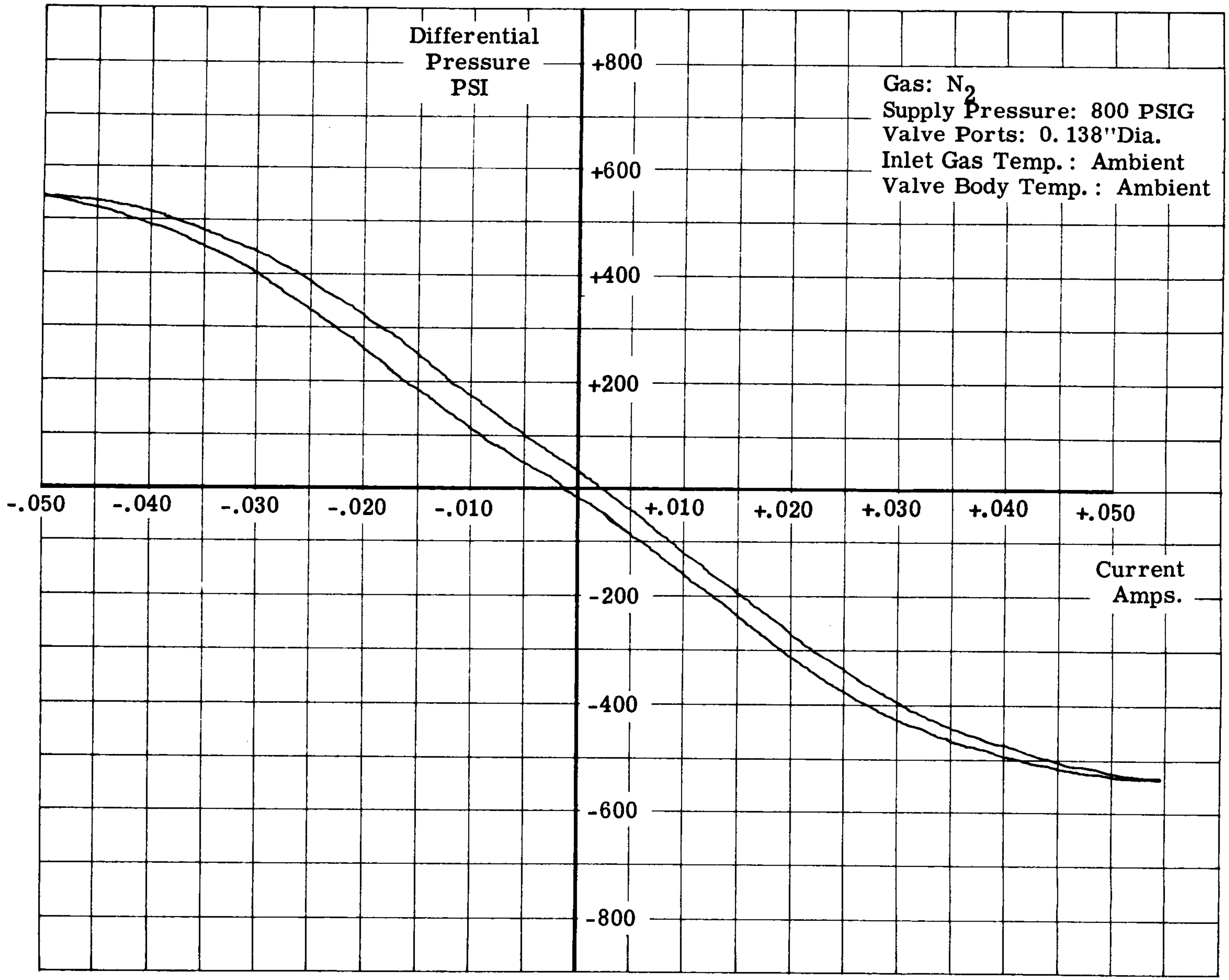


Figure 4-16. Motor Port Pressure Sensitivity (0.138" Diameter Ports).

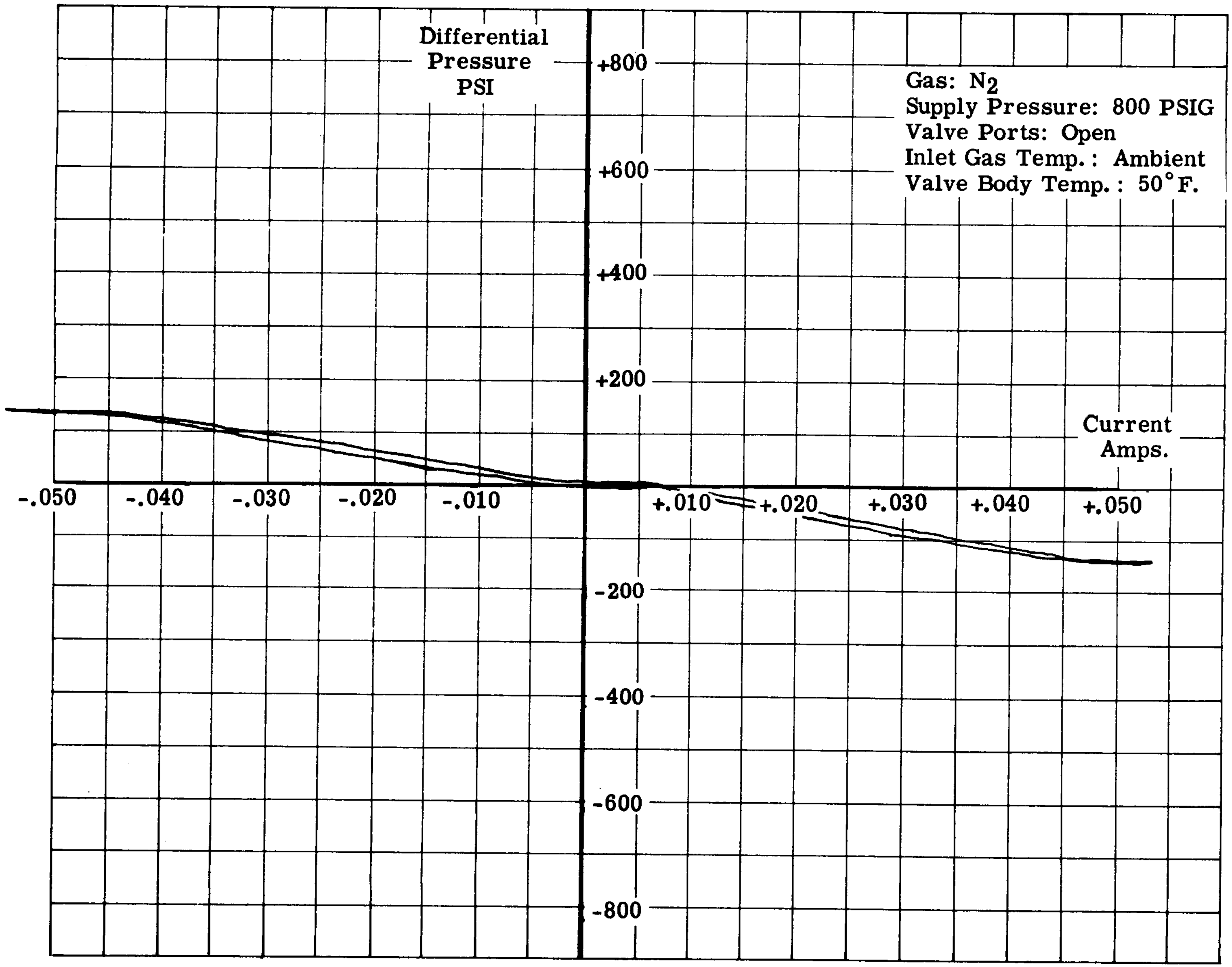


Figure 4-17. Motor Port Pressure Sensitivity (Open Ports).

Figures 4-14 to 4-17 show the effect of the valve load upon pressure gain as evidenced by the decreasing slope from the blocked motor port condition to the open motor port condition. The curve with the steep slope shown for the blocked port condition indicates valve hysteresis to be approximately 3.5ma or 3-1/2%. The amount of slope of the blocked port condition indicates the cutoff characteristics of the valve.

Spool friction and flow forces can be determined from Figure 4-18. The difference in pressure between the open port and blocked port curve indicates the effective flow force. The 10 psi pressure differential indicates a maximum flow force of approximately three pounds. Frictional forces are indicated as hysteresis and these appear very small.

The flow characteristics of the valve are shown in Figure 4-19 for various load orifices. The plotted scale is nonlinear due to the nonlinear characteristic of the transducer. The effect of valve load upon the flow gain is clearly demonstrated by the changes in slope for the different orifices.

4.2.3.3 Pilot Stage

The pilot stage performance is shown in Figure 4-20. Maximum available differential pressure, a measure of the ability of the spool to shear contamination, for the valve was initially 550 to 600 psi, but was later reduced to approximately 300 psi to improve valve stability. Normally only sufficient differential pressure to overcome friction and flow forces (approximately 12 psi differential pressure, see Figure 4-18, for this valve) is all that is required to shift the spool. Comparable servo valves generate only 60 to 110 psi differential pressure. The fact that the valve never stuck, considering the wear products that passed through the valve when the valve powered the motor during motor material wear tests, is attributed to the large available differential pressure generated by the pilot stage.

Some valves have demonstrated a statically unstable phenomenon arising from manifold losses due to valve flow. Since manifold losses are a function of spool position, the spool was mechanically locked at null and at approximately 80% stroke and the effects of spool position on pilot stage performance was observed. No significant change in pilot stage performance between null, or zero manifold loss, and the 80% stroke position was noted.

4.2.3.4 Stability Investigation

The stability plot shown in Figure 4-21 is a result of tests using nitrogen and hydrogen at room temperature with the high gain pilot stage. With nitrogen, a 0.040 damping chamber orifice, and the damping chamber length set at 1-1/2 inches, limit cycle oscillations will occur between 50 psig and approximately 190 psig supply pressure. Above 190 psig the valve is stable on nitrogen. With hydrogen, a 0.015 inch damping chamber orifice and the damping chamber length set at 1-1/2 inches, limit cycle oscillations occur between 50 psig supply pressure and 600 psig supply pressure. The shaded regions of the plot are regions of unacceptable limit cycle oscillation. Note the shaded portion of the hydrogen plot contains the nitrogen region. Using hydrogen, as

(Text continued on page 4-27)

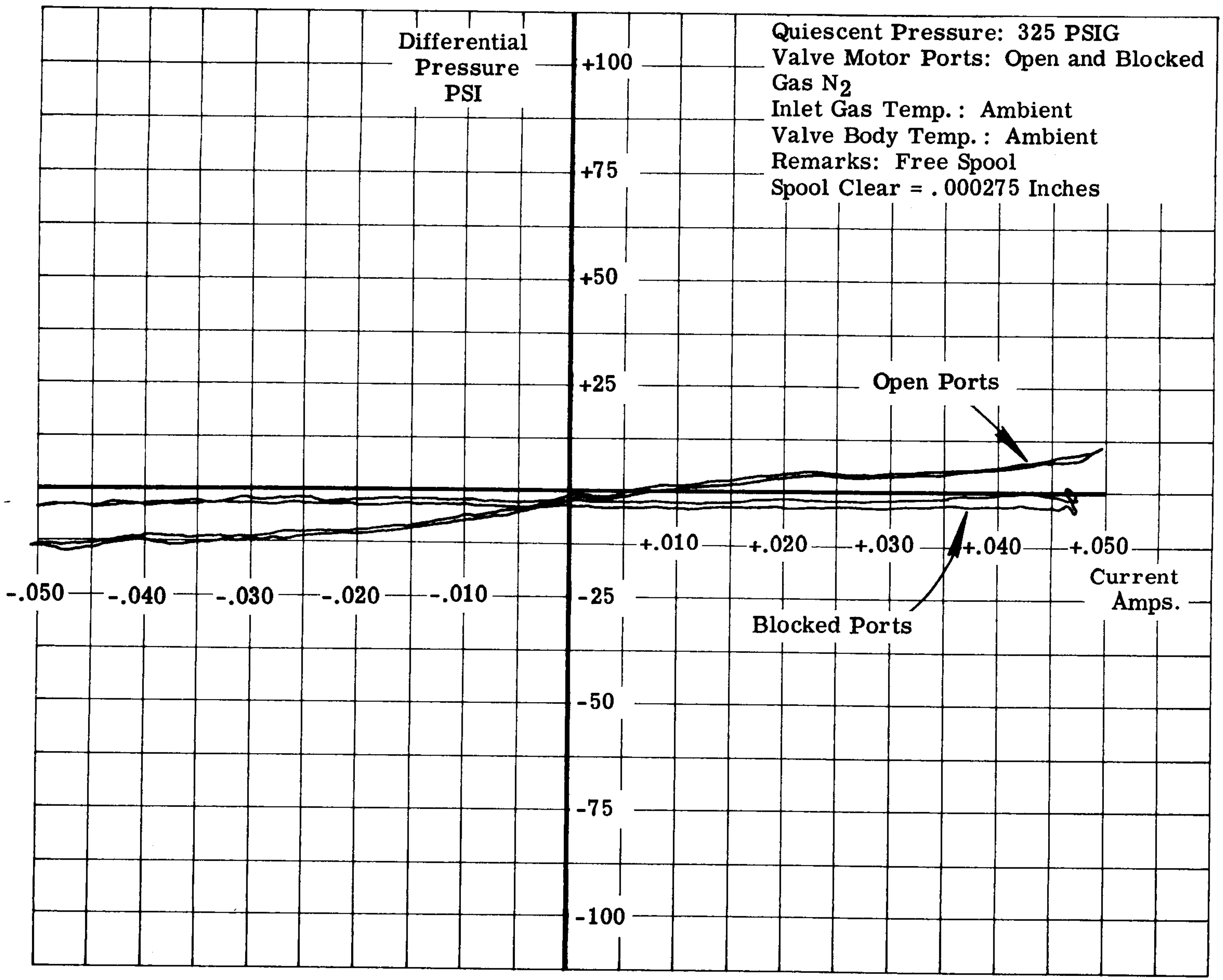


Figure 4-18. Pilot Stage Pressure Sensitivity.

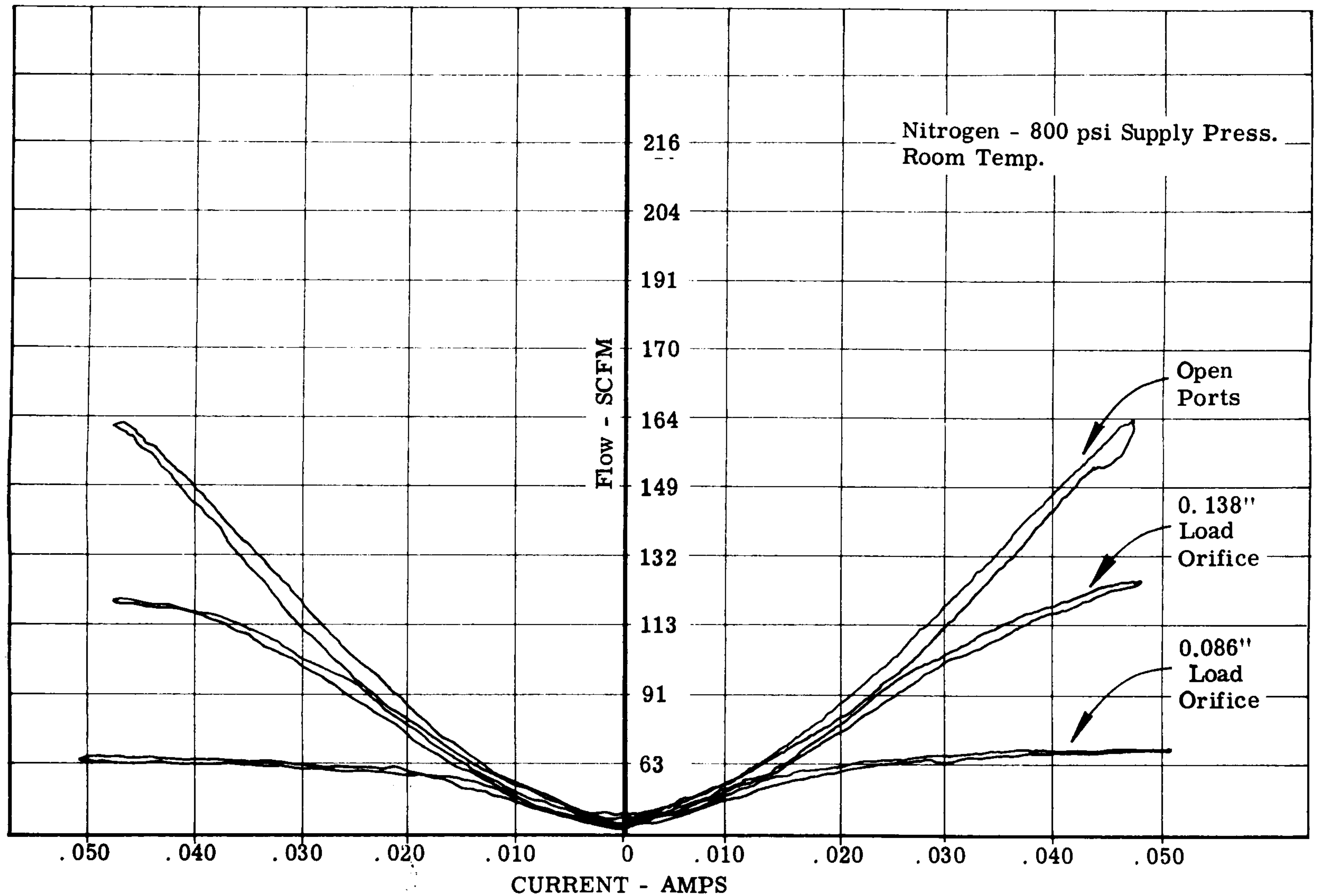


Figure 4-19. Flow Characteristics.

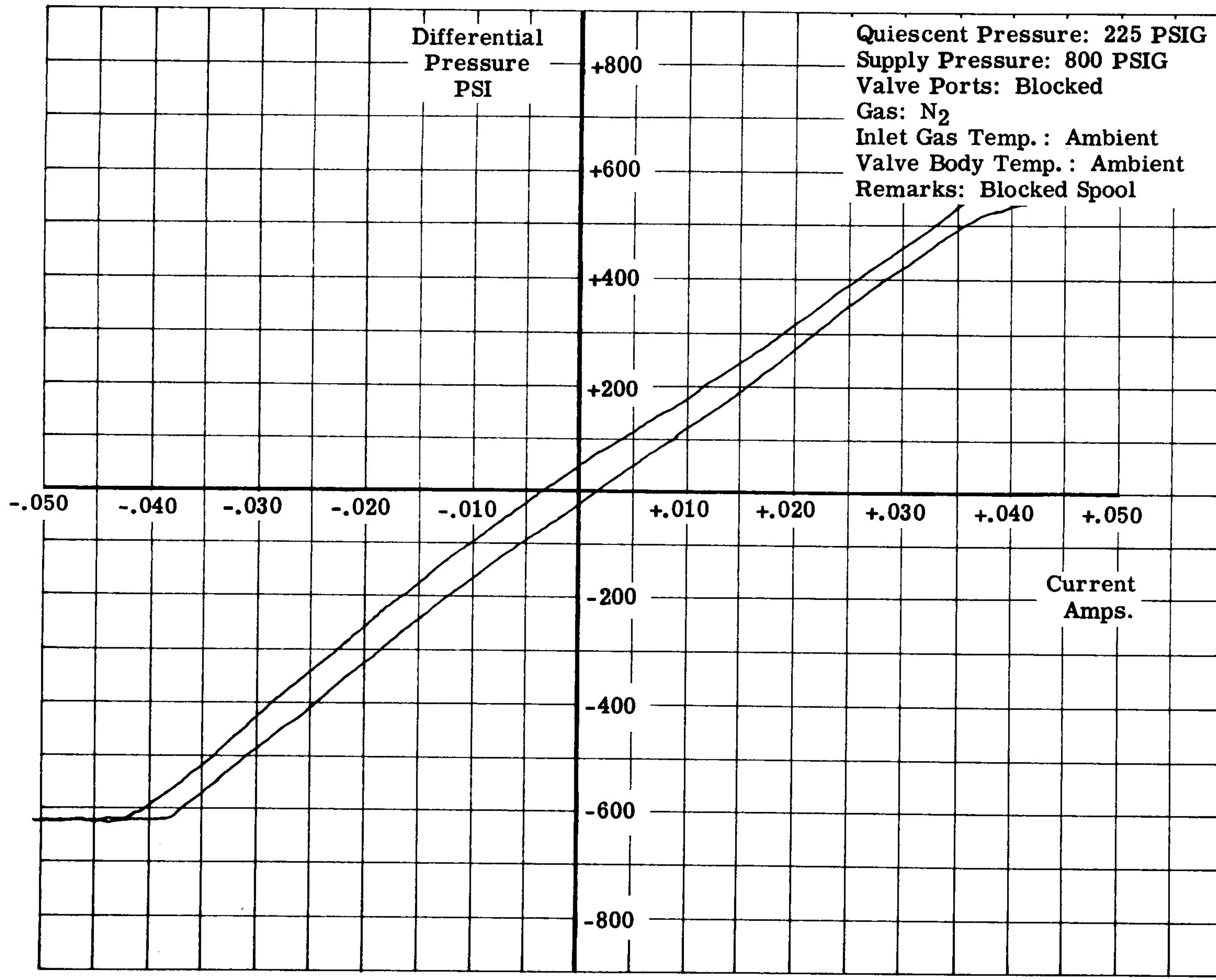


Figure 4-20. Pilot Stage Pressure Sensitivity.

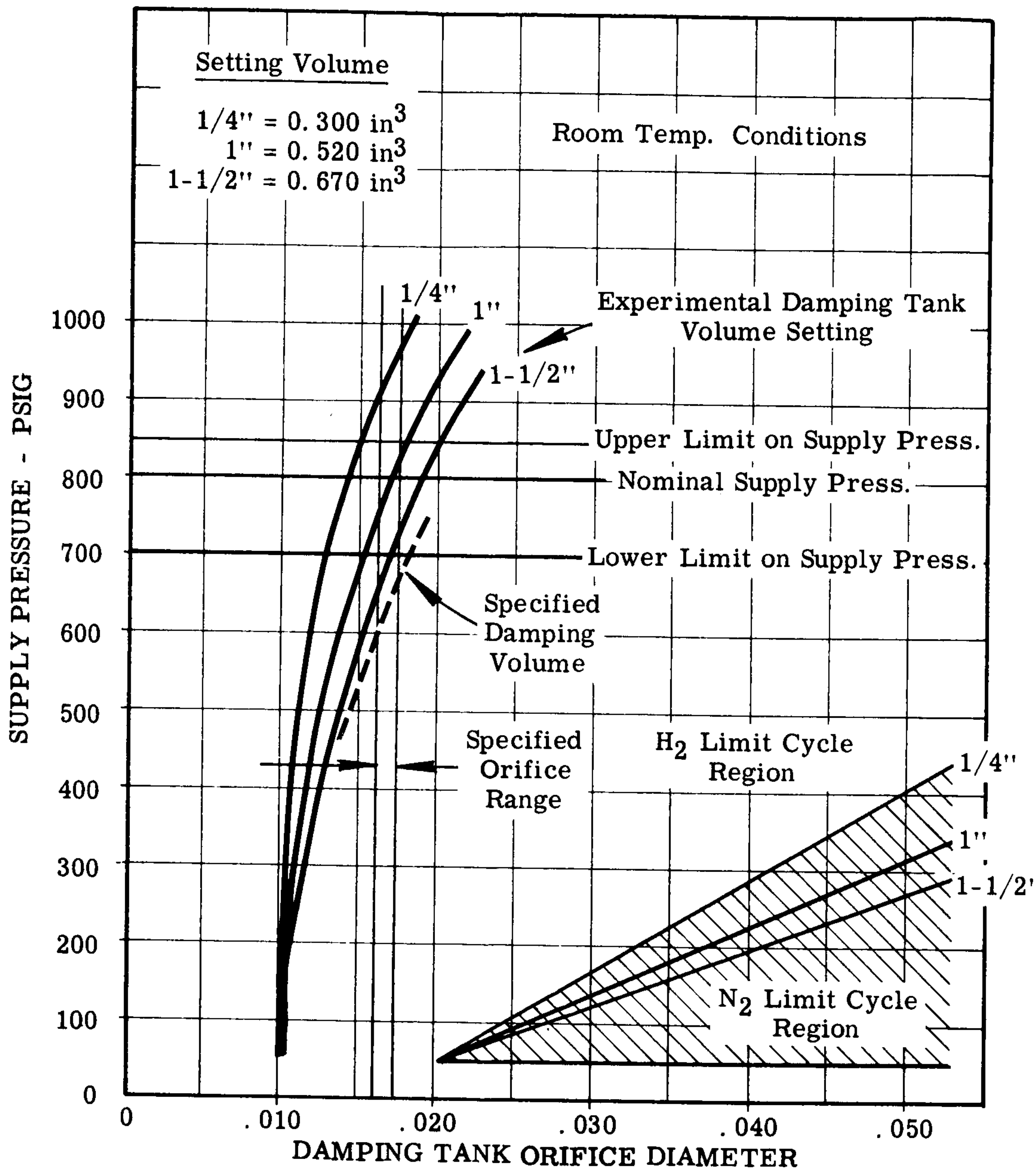


Figure 4-21. Stability Plot.

supply gas temperature is lowered, the upper border of the hydrogen plot is expected to move toward the upper border of the nitrogen limit cycle region. Therefore, if the valve is stable on hydrogen at room temperature, it will be stable at lower temperatures using hydrogen.

The damping volume of the servo valve is shown by the dashed line. The range of the orifice diameter specified in the design is also shown as well as the nominal and expected supply pressure range.

4.2.3.5 Frequency Response

Figures 4-22 and 4-23 show the frequency response of the servo valve using room temperature, 800 psig, nitrogen with the Midwestern and Servotronics torque motors. The frequency response is the ratio of the spool position to the current input. The spool position was measured with a Bently proximity transducer mounted in the experimental end caps. The effect of the two torque motors on valve performance is clearly demonstrated.

4.2.4 Servo Motor Development Tests

The purpose of these development tests was to determine the characteristics and durability of the breadboard motor design as related to the final design.

4.2.4.1 Description of Parts Tested

The breadboard and final motors were identical with respect to displacement, rotor, and vane configurations. The prime differences were in external configuration. The final design incorporated the interconnecting motor pressure passages within the two-piece housing rather than using external transfer tubes,and the rear cap was sealed to prevent leakage to atmosphere. Also, the delivered configuration contains 0.015 inch Rulon sheets cemented to the end caps to prevent damage to the rotor ends and end caps. The final delivered motor parts are covered by Parts List 2775037 included in Appendix A.

Operational characteristics of the two motors should have been identical except for differing characteristics due to different vane materials. The vanes in S/N 3 motor (delivered in Actuator S/N 1) were made from Du Pont's SP-2 material and S/N 2 motor (delivered in Actuator S/N 2) used vanes made from Rulon "A".

4.2.4.2 Test Equipment and Method of Test

Development motor tests were run both at the Bendix Research Laboratories Division and Bendix Products Aerospace Division. The test equipment and methods were somewhat different; therefore, both are described.

4.2.4.2.1 Bendix Research Laboratories Division

1. All tests were performed using nitrogen gas at approximately 40° F.

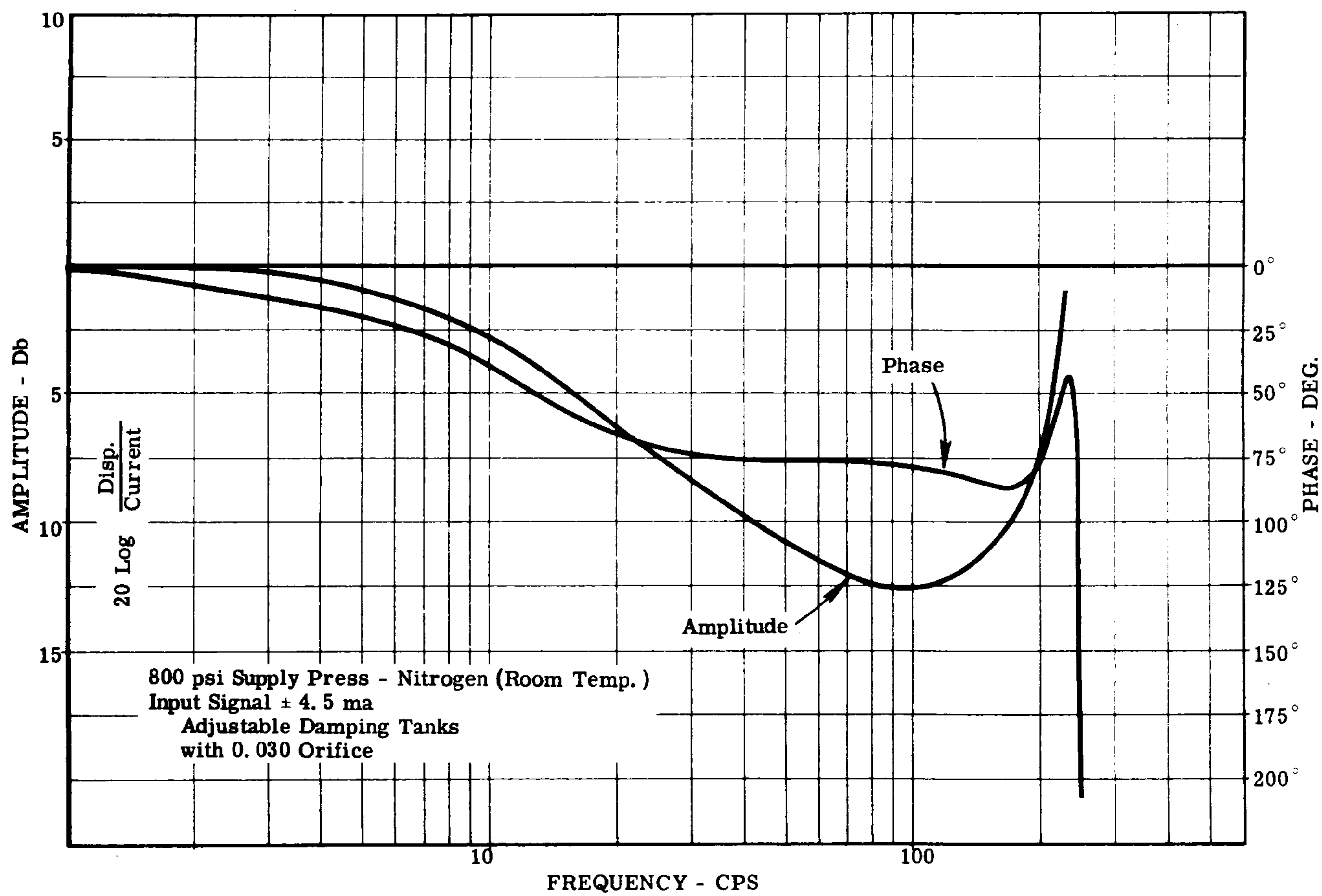


Figure 4-22. Servo Valve Frequency Response With The Midwestern Torque Motor.

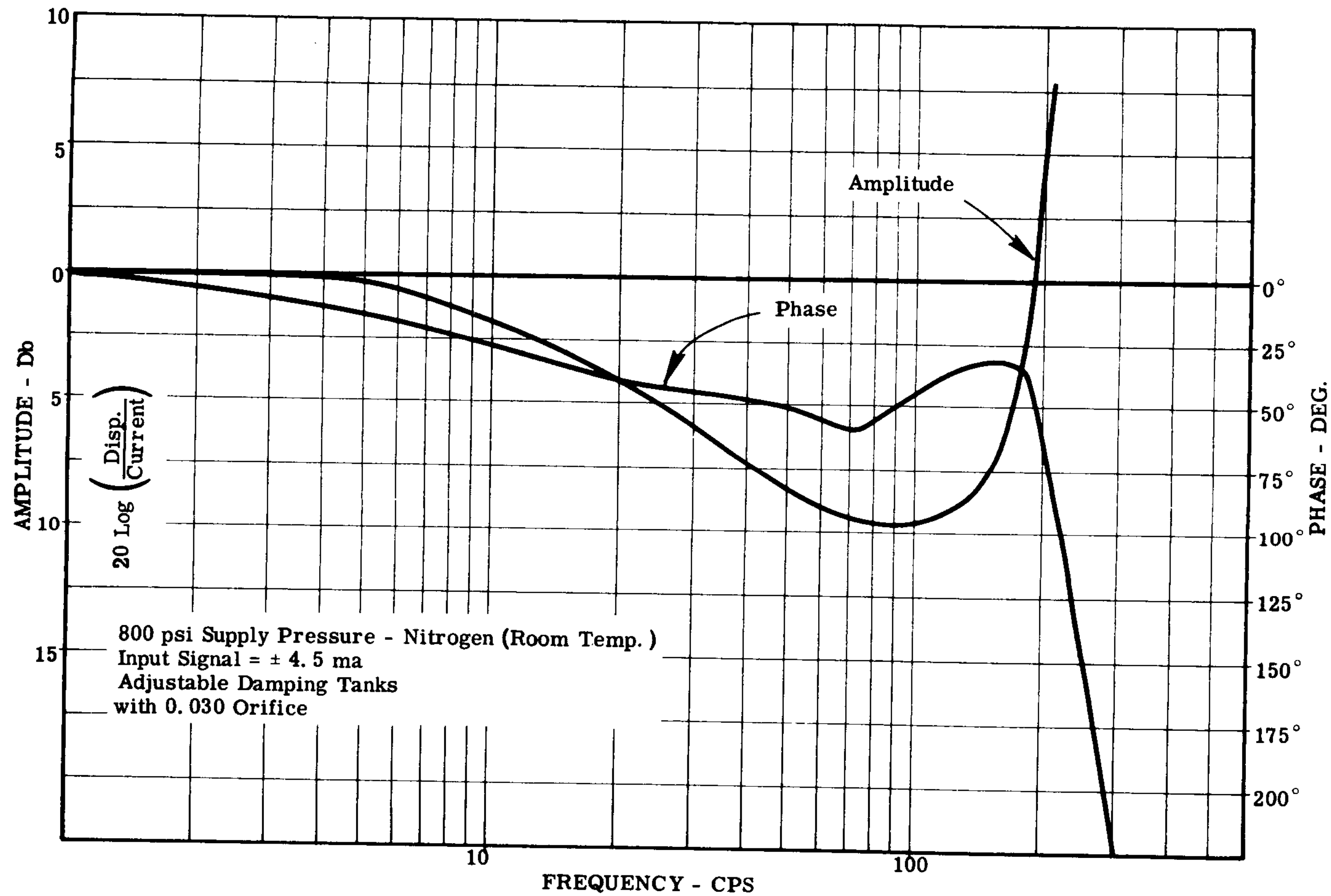


Figure 4-23. Servo Valve Frequency Response With The Servotronics Torque Motor.

2. Figure 4-24 is a schematic diagram of the test setup.

3. An 8-channel Sanborn Model 850 Recorder was used to simultaneously record the following information: Input signal, speed, flow, torque, supply pressure, P_1, and P_2.

4. The motor was loaded with a Vickers hydraulic pump. The load was varied by changing an orifice on the pressure side of the pump.

5. The motors were tested for performance by varying the load in increments for given values of valve stroke. Continuous recordings of all data were made on the Sanborn Recorder.

4.2.4.2.2 Bendix Products Aerospace Division

The motor was mounted on a test fixture as shown in Figure 4-25 which contains a torque sensor and a magnetic brake so that output torque and speed can be recorded on an X-Y plotting board or read directly. In addition, a pressure pickup adaptor was installed between the servo valve and motor so that motor P_1 and P_2 pressures or ΔP across the motor could be recorded. An orifice type flowmeter was installed in the gaseous N_2 supply line to the motor.

The normal test run at Bendix Products Aerospace Division was a check of motor pressures, stall torque, and gas flow vs. torque motor current over a complete hysteresis loop of ±40ma. This permitted evaluation of ΔP gain, torque gain and hysteresis, and gas flow for comparison of the various motor and valve configurations. In addition, a check of ma deadband required to reverse the motor was taken for comparative evaluation.

4.2.4.3 Summary of Test Results

Figures 4-26 through 4-29 present torque-speed, horsepower, and specific fuel consumption on the breadboard motor. It will be noted that successive runs show a marked improvement in torque-speed characteristic through Figure 4-28. However, there was considerable drop off on run #4, Figure 4-29. The vanes used in the prototype motor were made from DuPont's SP-1 material with dispersed MoS_2. It was found that these vanes required considerable run-in to obtain optimum performance and, if allowed to stand for any length of time, the characteristics would deteriorate and another run-in was necessary.

Figure 4-30 indicates horsepower and specific fuel consumption vs. speed for servo motor S/N-3 (servo valve No. 1) at torque motor currents of 20, 30, and 50 ma with nitrogen gas as the servo fluid.

Figure 4-31 indicates torque vs. speed at various torque motor currents for the same motor-valve combination. Figure 4-32 presents no-load speed vs. torque motor currents.

(Text continued on page 4-40)

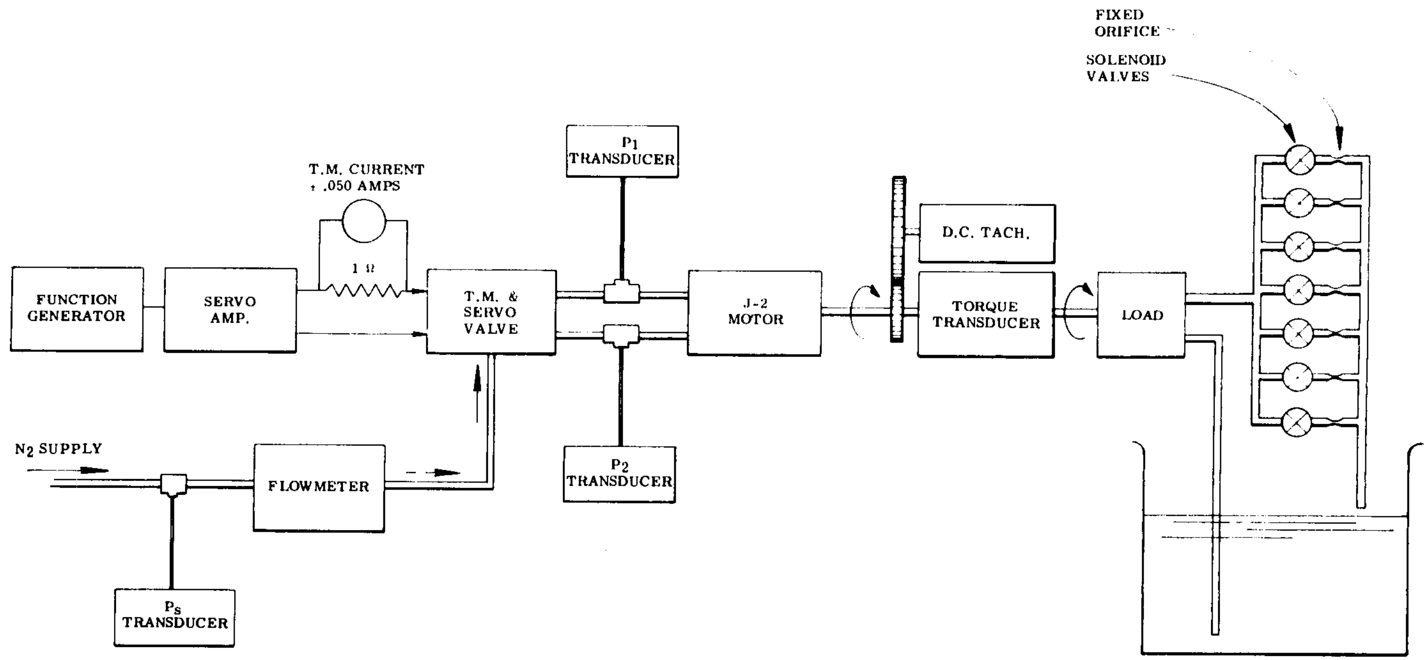


Figure 4-24. Schematic Diagram Of Bendix R.L.D. Servo Motor Test Setup.

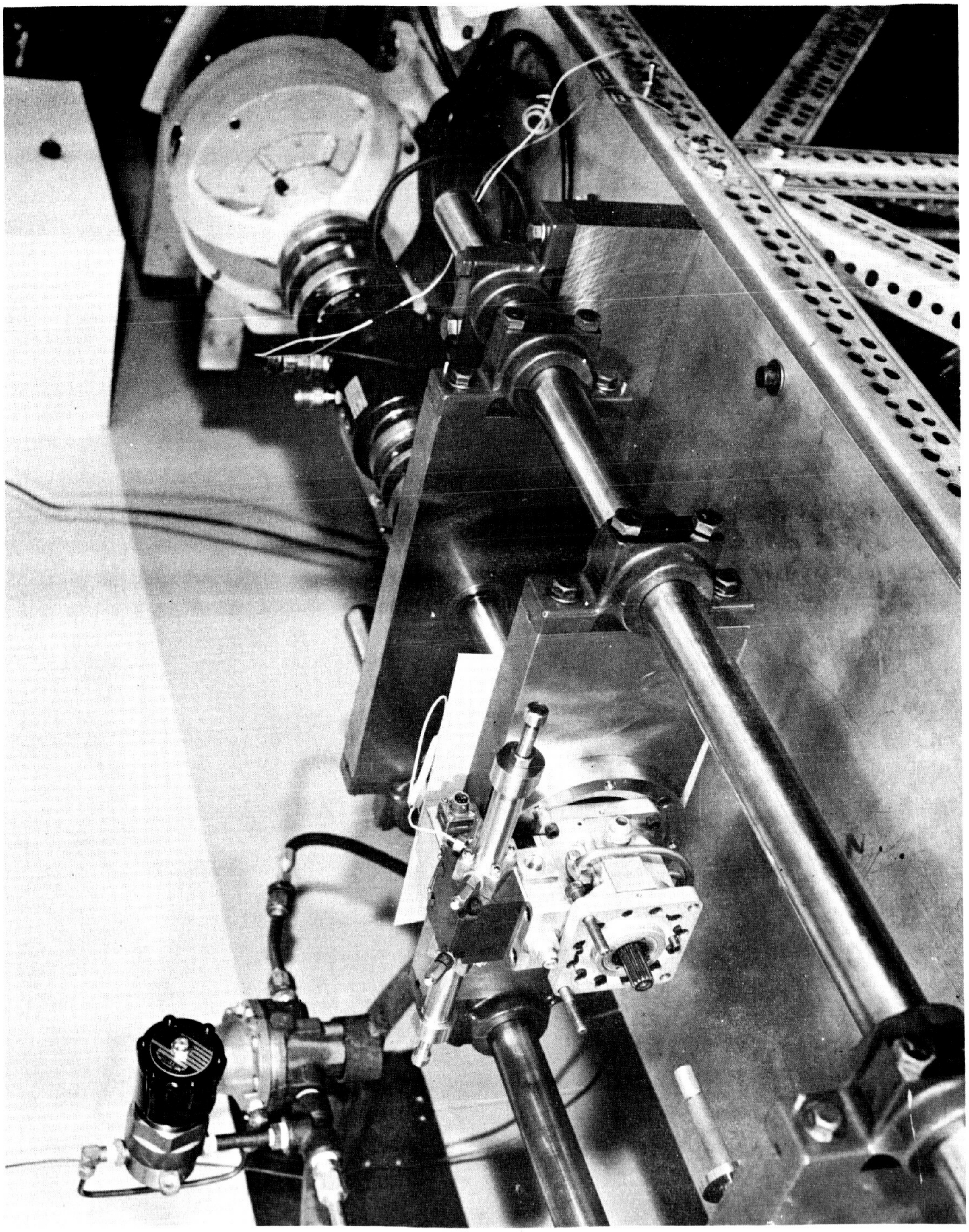


Figure 4-25. Bendix P.A.D. Servo Motor Test Setup.

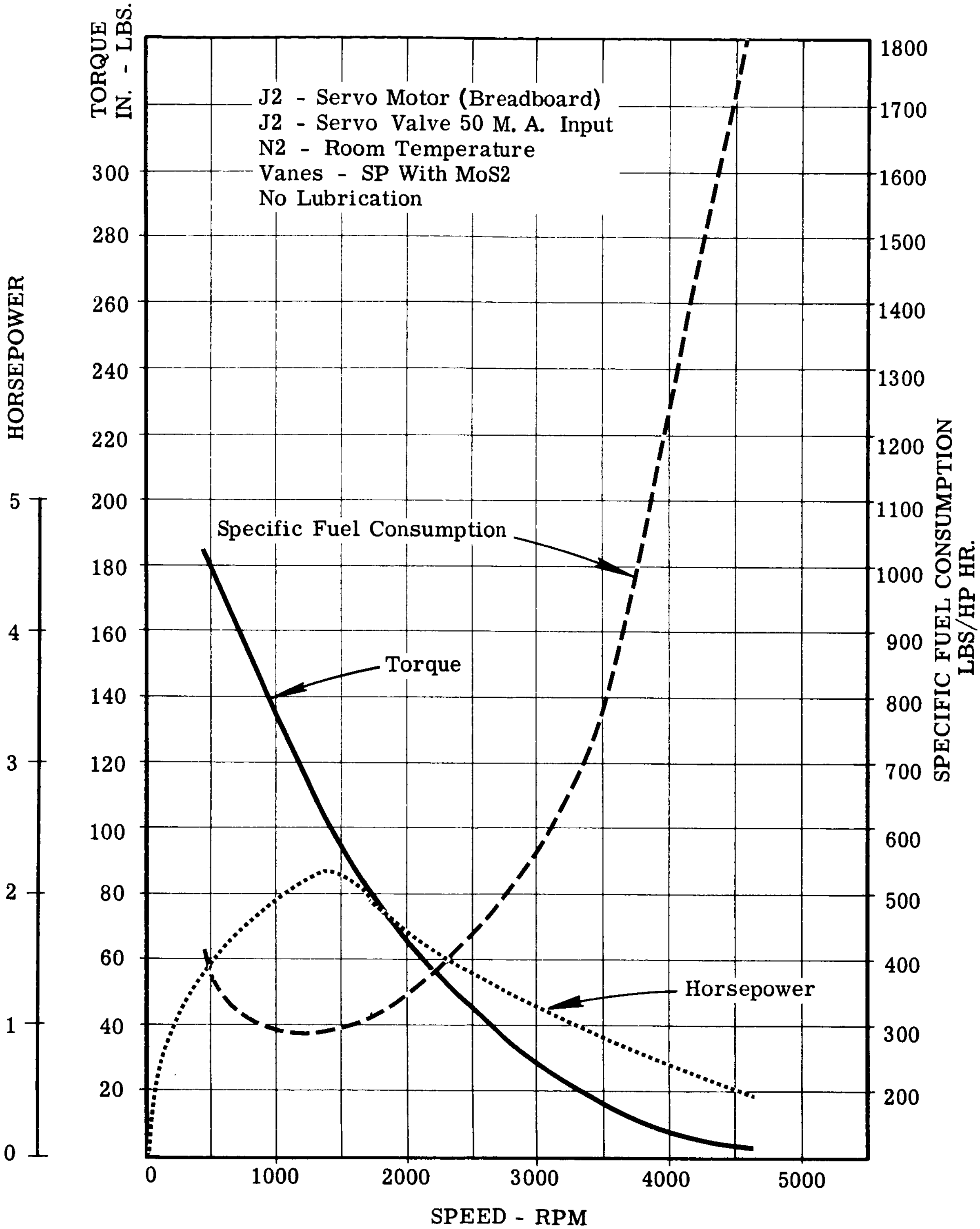


Figure 4-26. Breadboard Servo Motor Characteristics - First Run.

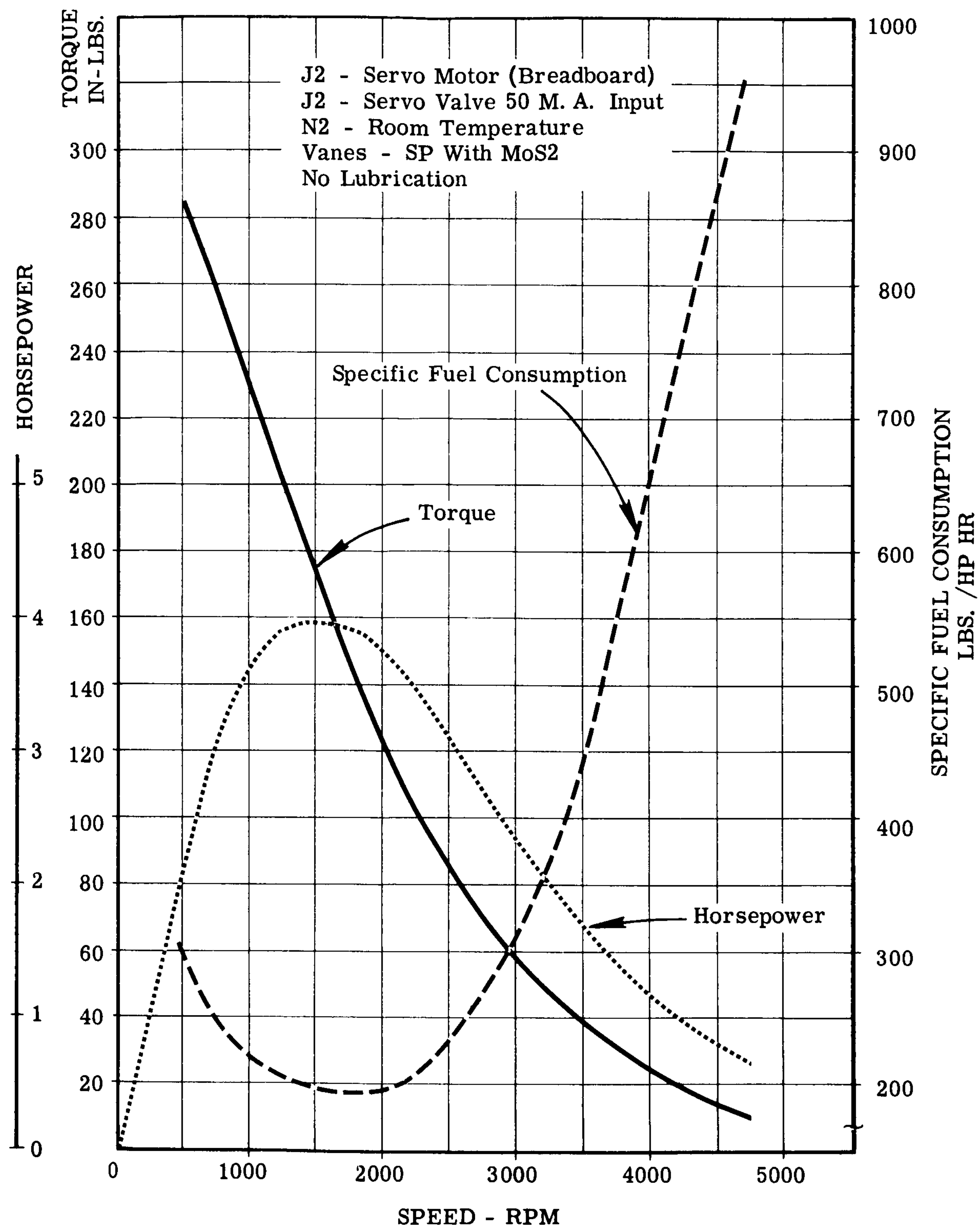


Figure 4-27. Breadboard Servo Motor Characteristics - Second Run.

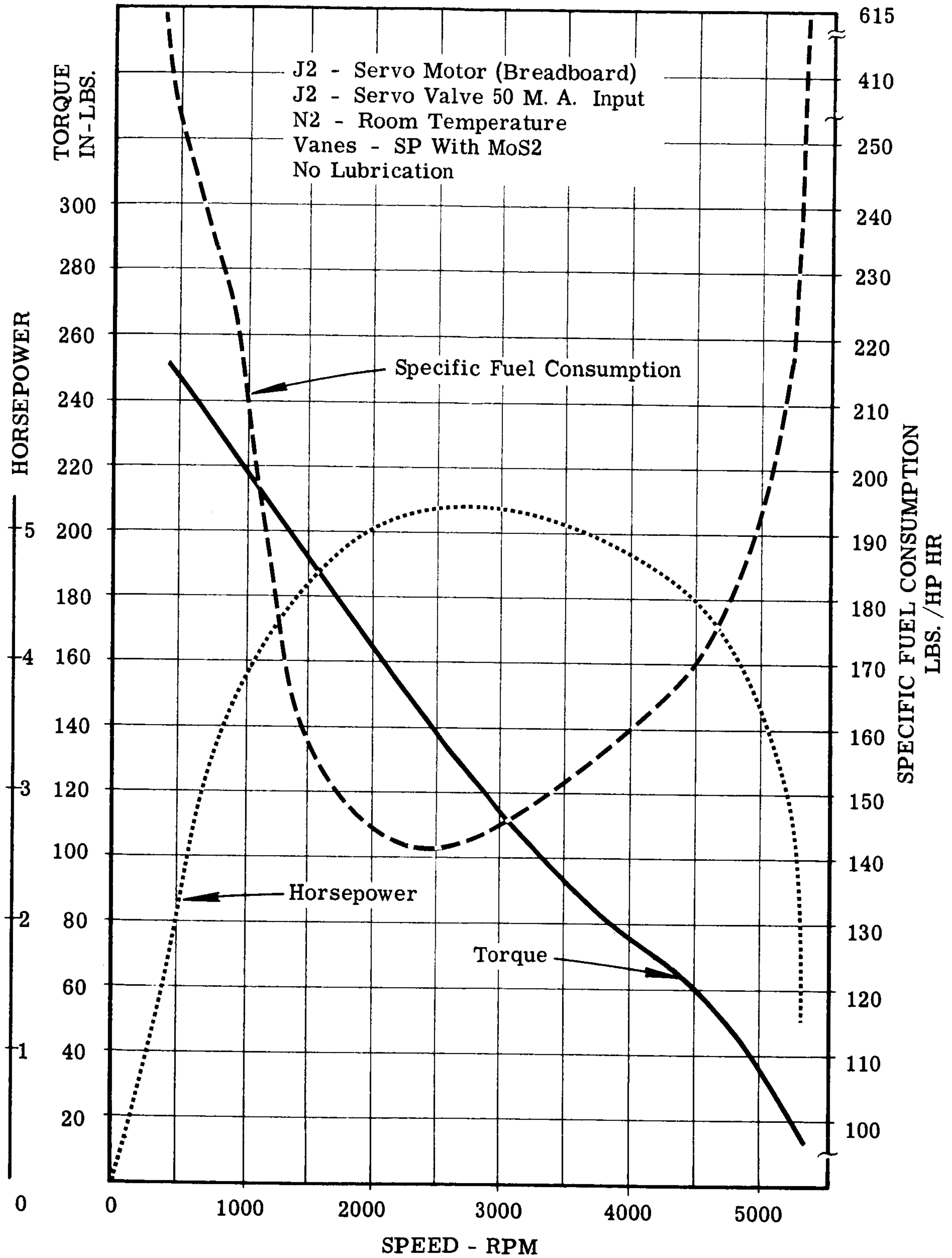


Figure 4-28. Breadboard Servo Motor Characteristics - Third Run.

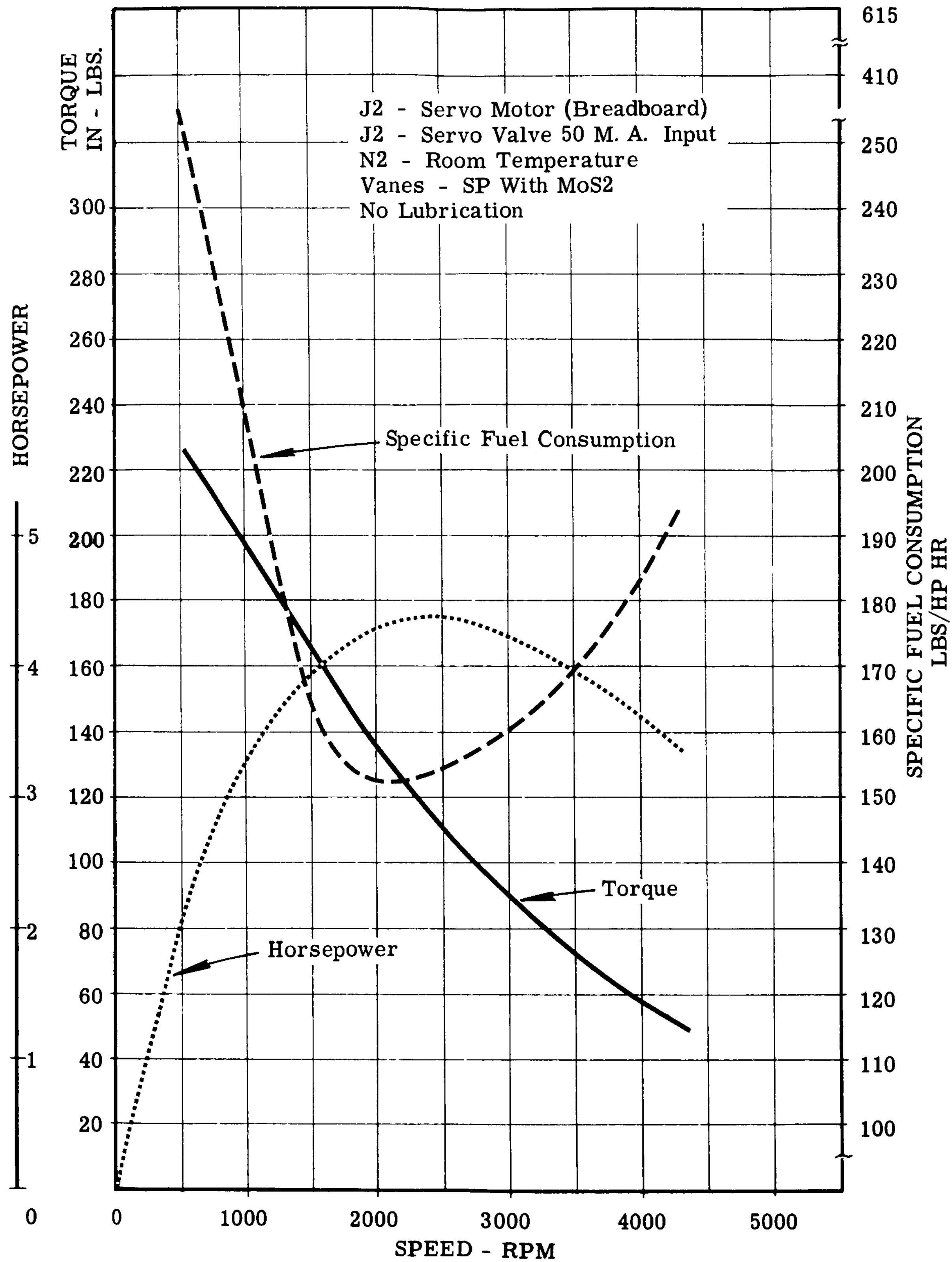


Figure 4-29. Breadboard Servo Motor Characteristics - Fourth Run.

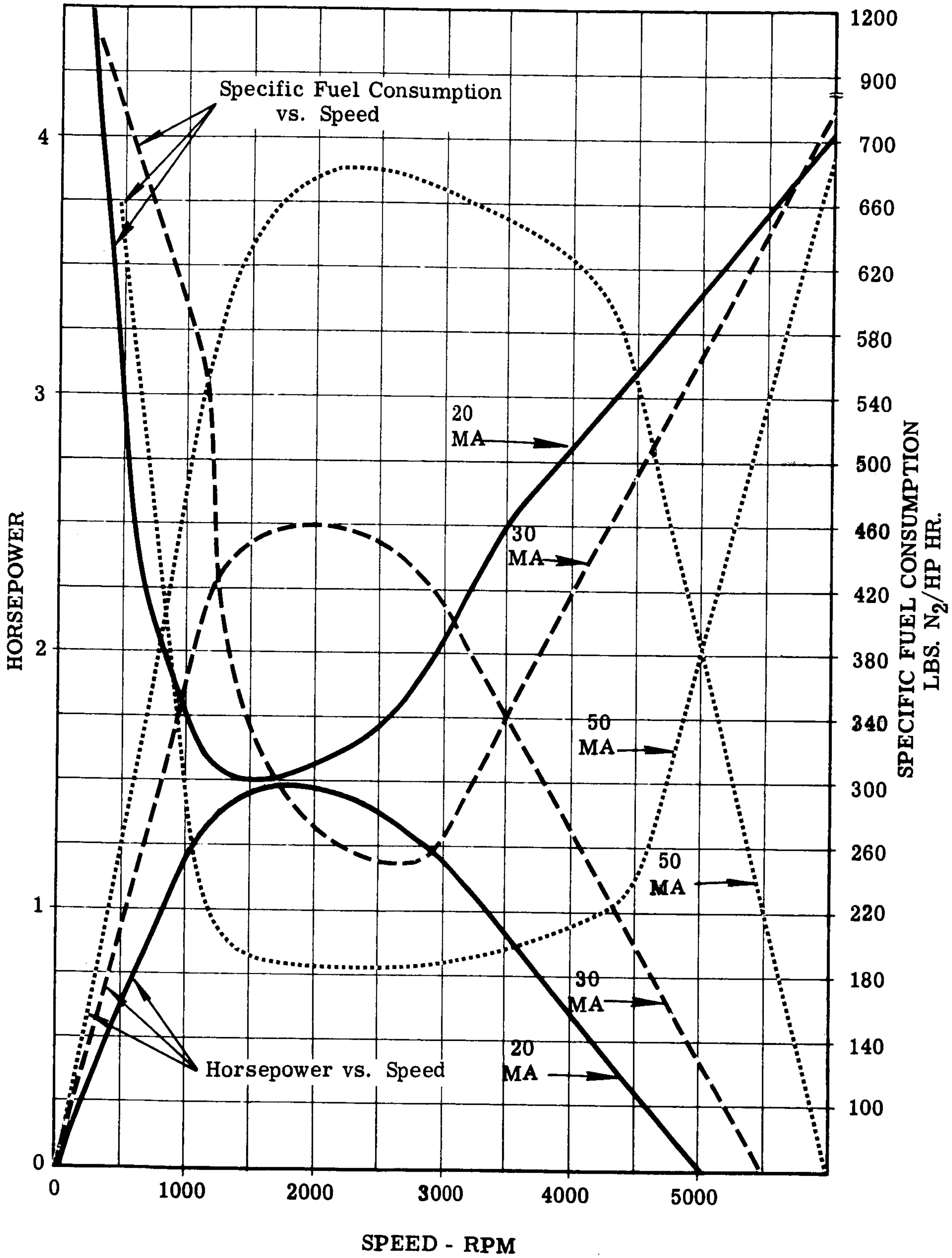


Figure 4-30. Prototype Servo Motor Characteristics.

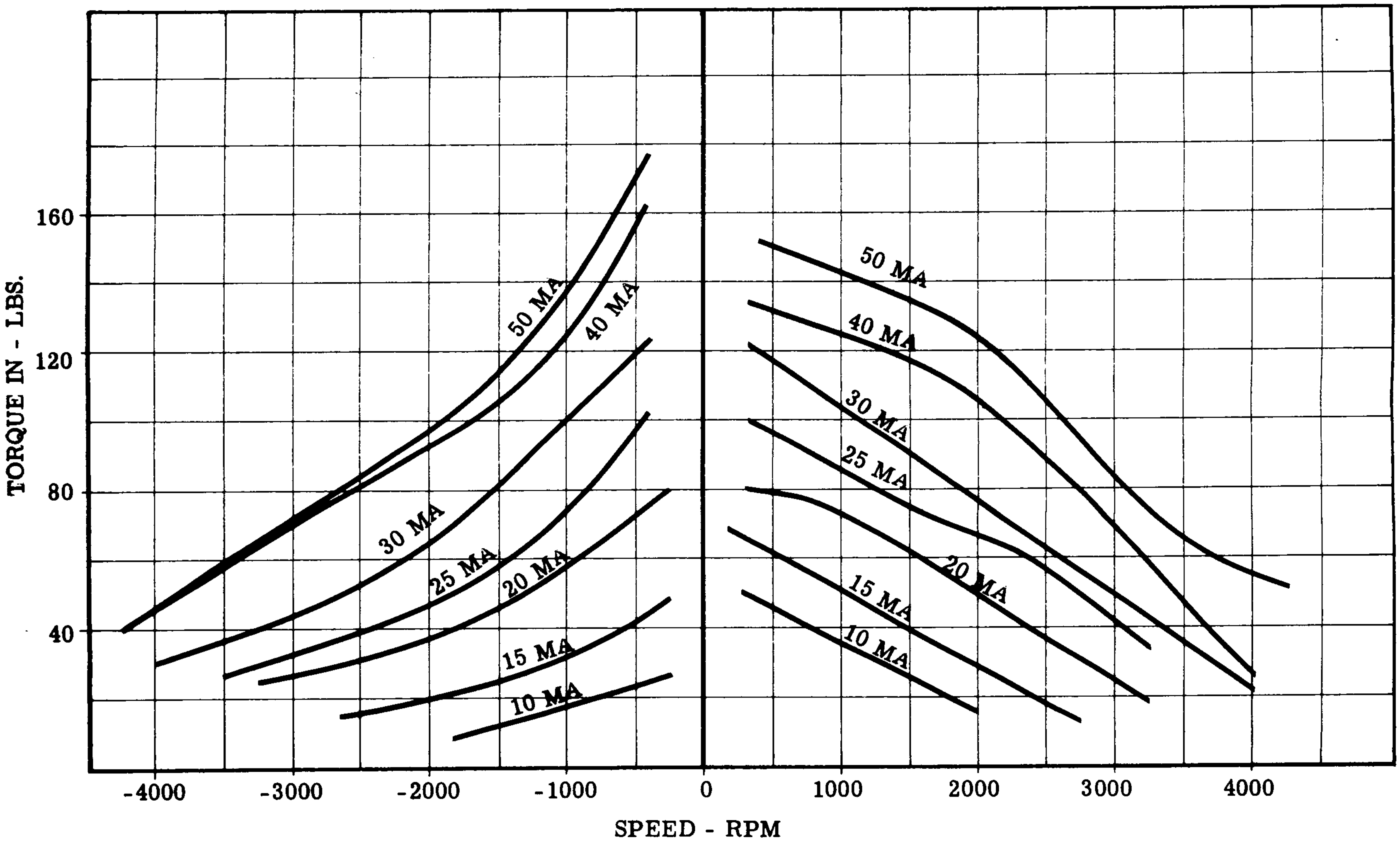


Figure 4-31. Torque vs. Speed For Prototype Servo Motor.

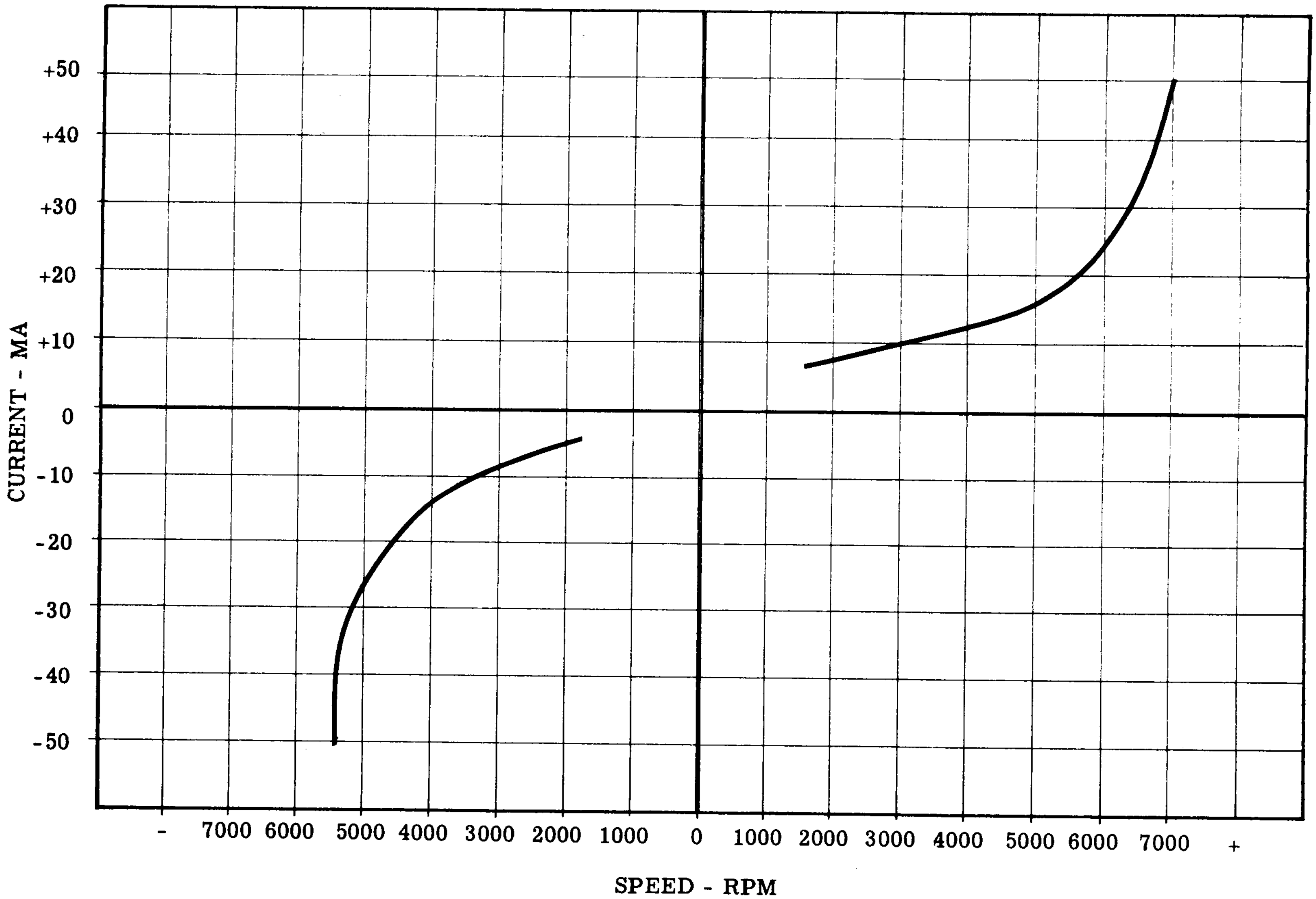


Figure 4-32. Current vs. No-Load Speed For Prototype Servo Motor.

The frequency response of the prototype motor was measured by comparing motor speed amplitudes at various input signal frequencies to a base established at 0.05cps with an amplitude of ±10ma. Plots of the amplitude ratio and phase shift are shown in Figure 4-33.

The effect of various vane materials and lubricants upon motor output power was investigated with the breadboard motor. Results are presented in Figures 4-34 and 4-35.

The fuel consumption for the prototype motor was determined as a function of current to the valve. Results are shown in Figure 4-36.

The final tests conducted at Bendix Products Aerospace Division on the prototype motor as shipped on actuator S/N 1 are shown in Figures 4-37 and 4-38. Motor S/N 3 and servo valve S/N 1 were shipped on this unit with Midwestern torque motor S/N 102. This motor used SP-2 tapered vanes. It should be noted that these vanes, if allowed to stand for a short period, will on the first check of deadband, require additional input signal to reverse the motor. In this case, a total deadband of 11.5ma was required for cycling and, if the unit was allowed to stand for two hours, the first check of deadband required 22ma. After cycling three minutes, the deadband was reduced to 12.5ma.

The final tests conducted on S/N 2 servo motor with servo valve S/N 2 and torque motor S/N 101 as shipped on S/N 2 actuator are shown in Figures 4-39 and 4-40. This motor used vanes of Rulon "A" which do not contribute any additional deadband due to an inactive actuator. The deadband was, in this case, 12ma.

The following comparative data was obtained from the latter motor performance curves:

Item	S/N 3 Motor	S/N 2 Motor
Quiescent Flow GN_2	.026 pps	.05 pps
Saturation Flow	.174-.175 pps	.179 pps
Quiescent Pressure	80 psig	100 psig
Deadband (Torque 15 in/lbs)	11.5ma	12ma
ΔP To Rotate	53-68 psi	75-85 psi
Torque Gain Maximum	7000-7800 in. lb/amp.	8900-9600 in. lb/amp.
Saturation Torque (±40ma)	172-190 in. lbs.	188-231 in. lbs.
Torque Hysteresis (±10ma from null)	128-130 in. lbs.	80 in. lbs.
Saturation ΔP (±40ma)	520-540 psi	453-505 psi

(Text continued on page 4-49)

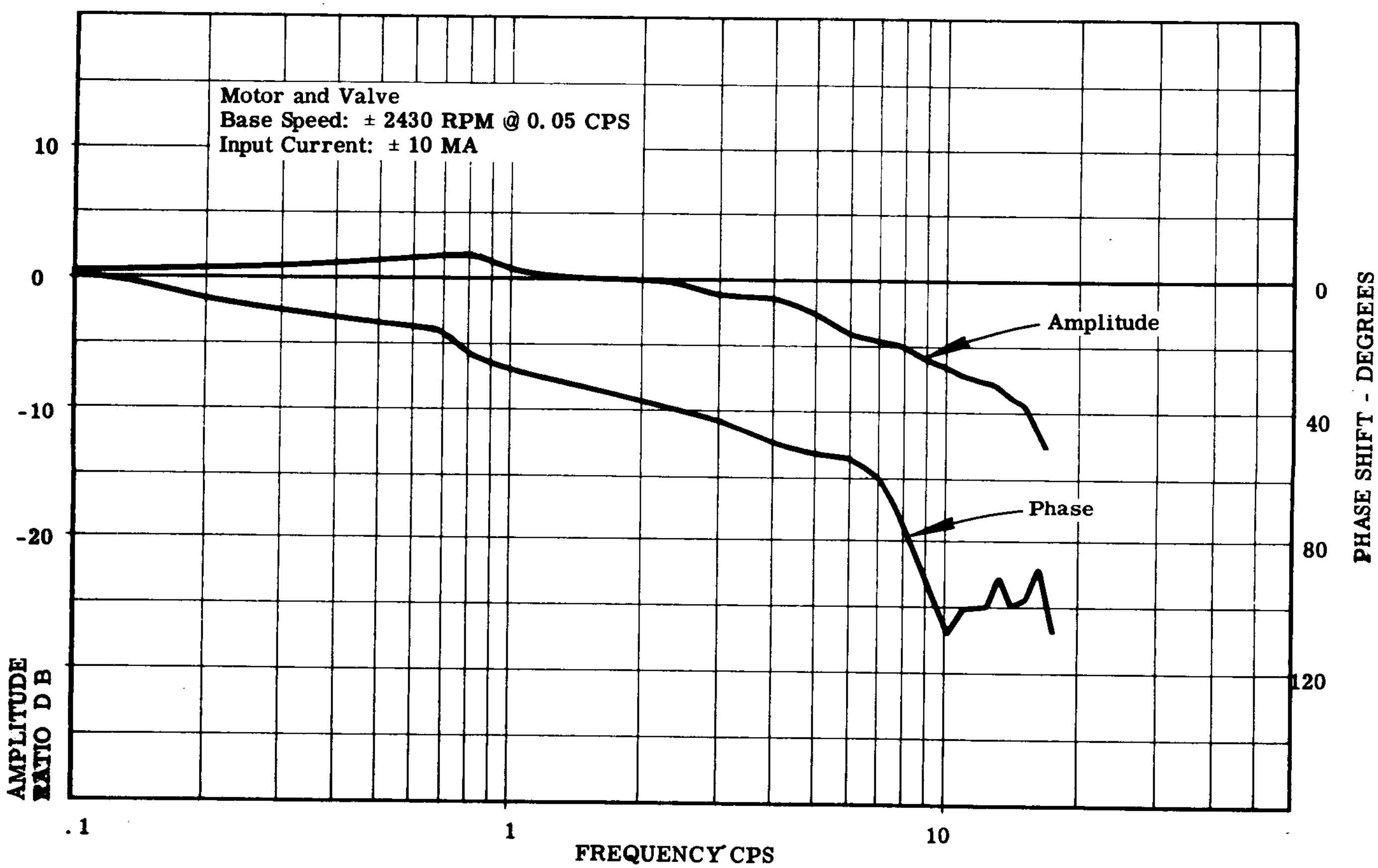


Figure 4-33. Frequency Response For Prototype Servo Motor.

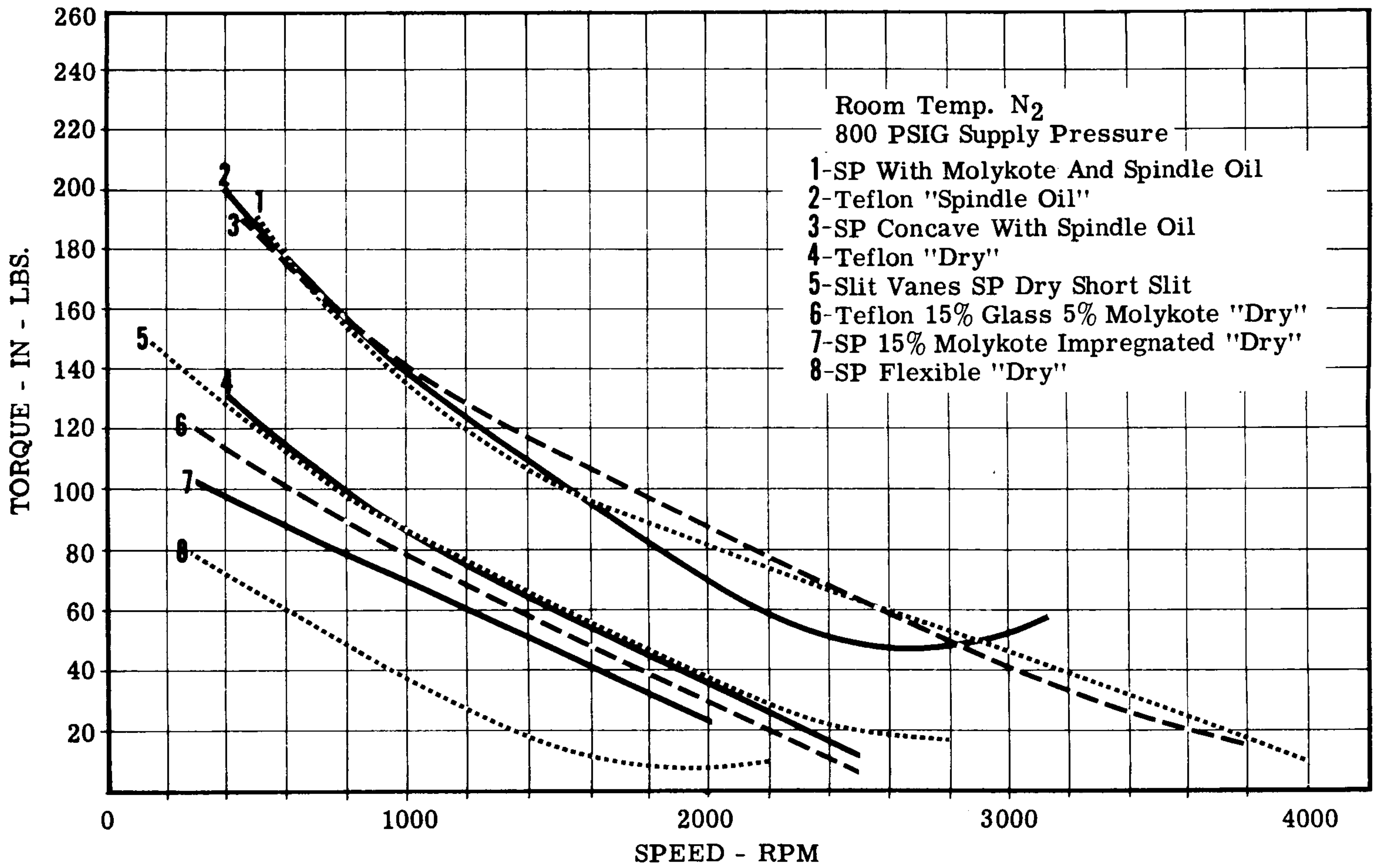


Figure 4-34. Torque vs. Speed Characteristics For Various Motor Vanes .

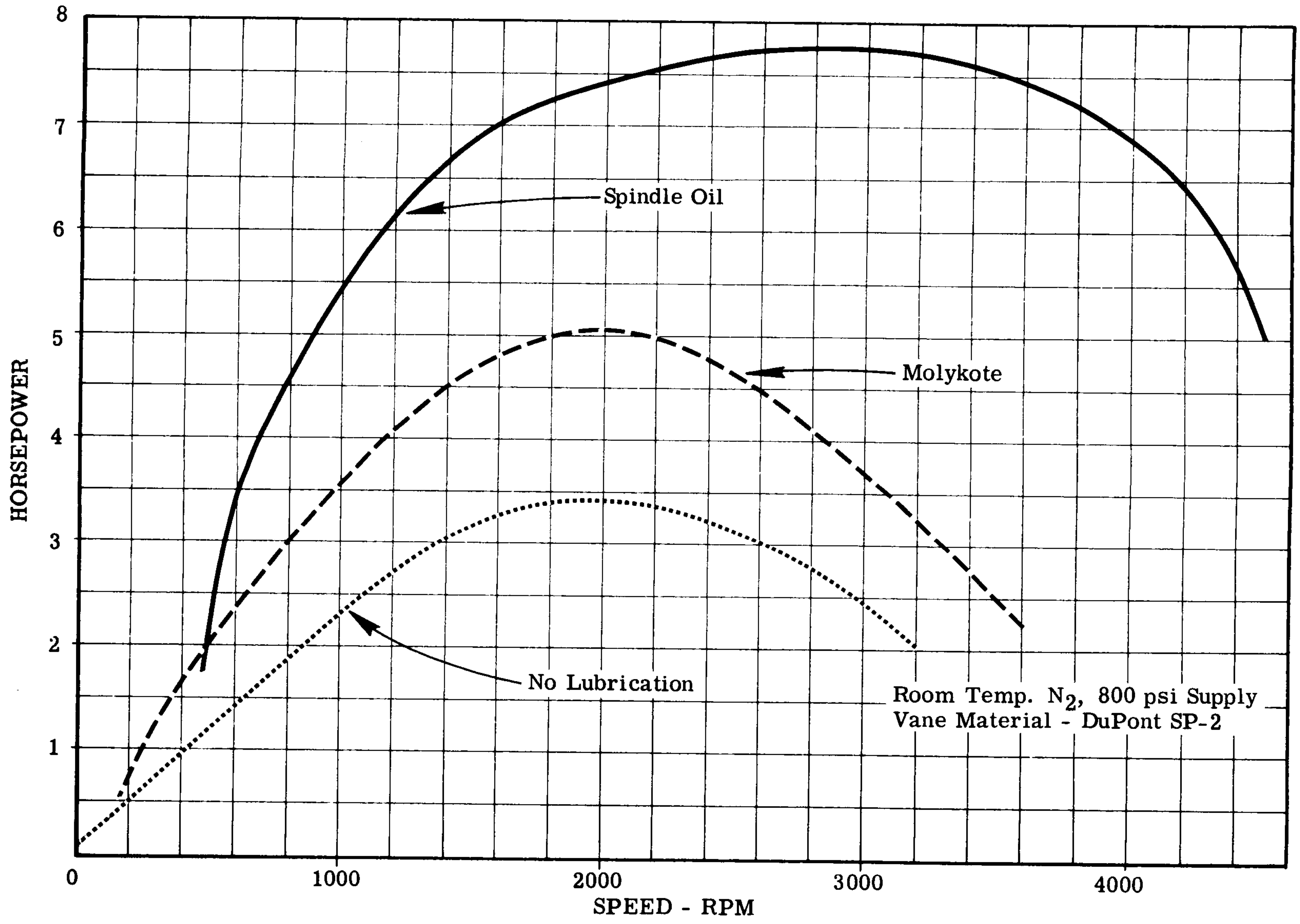


Figure 4-35. Horsepower vs. Speed Characteristics With Various Motor Lubricants.

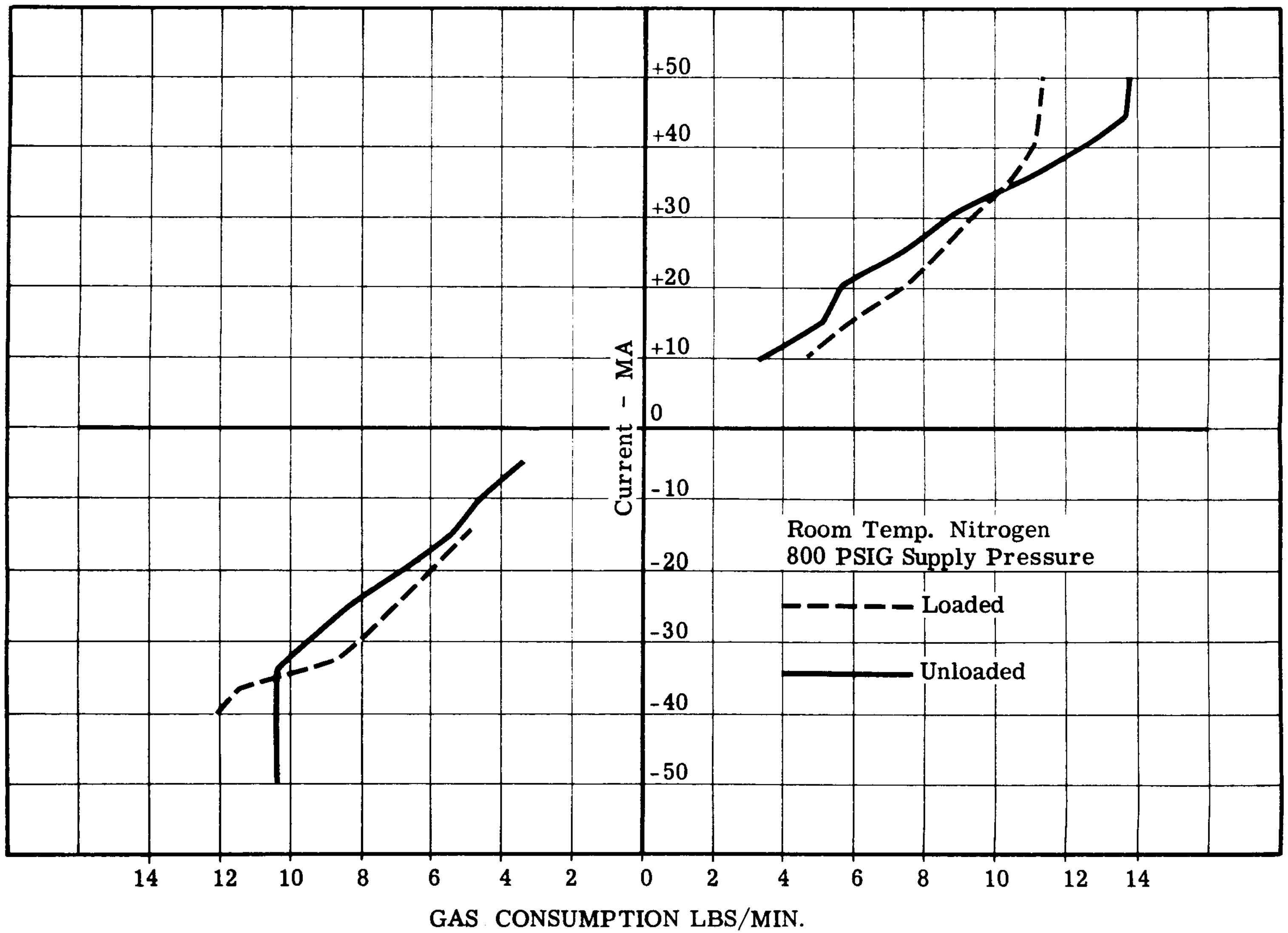


Figure 4-36. Input Current vs. Gas Consumption.

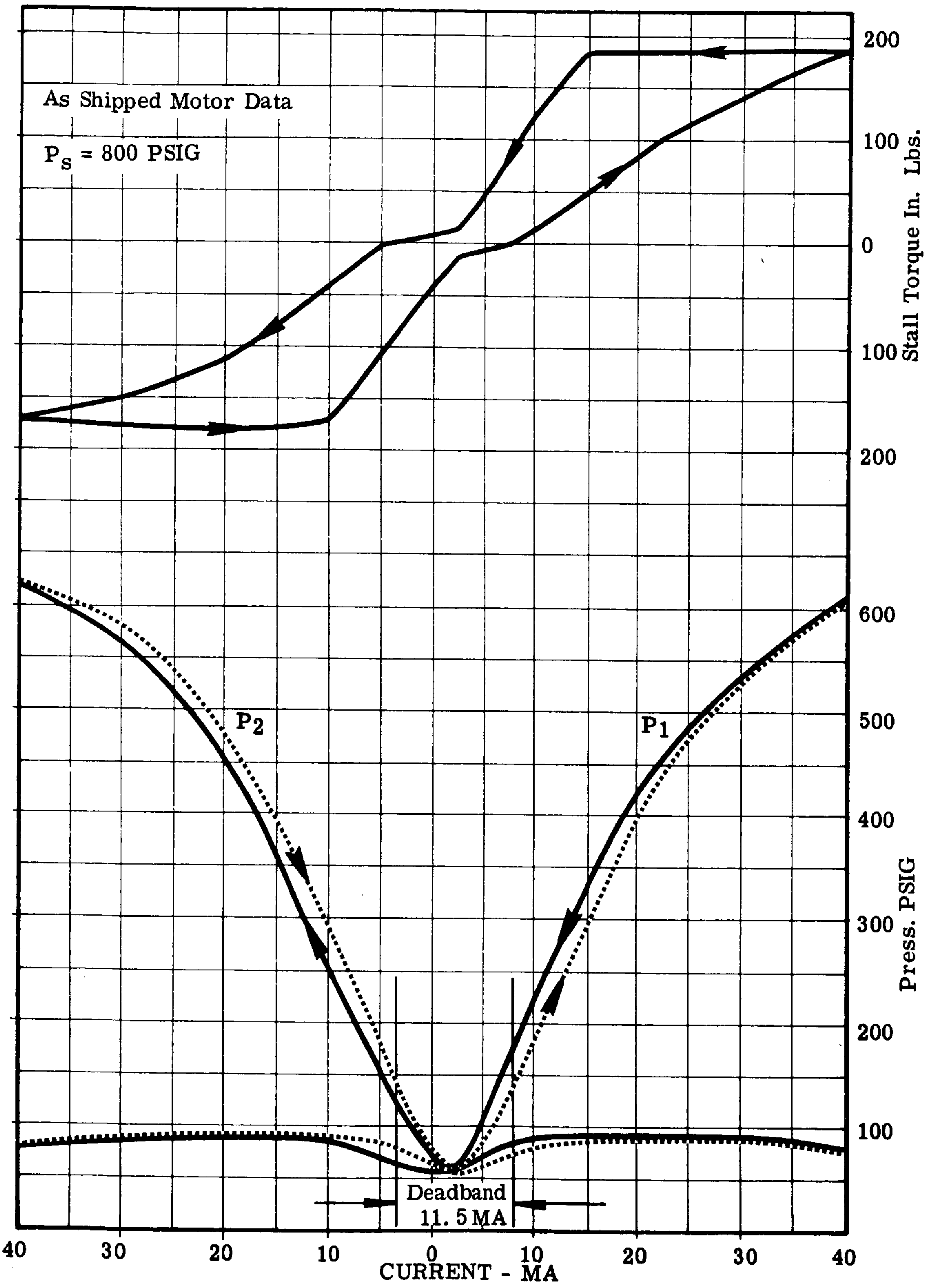


Figure 4-37. Stall Characteristics - Actuator S/N 1 Motor.

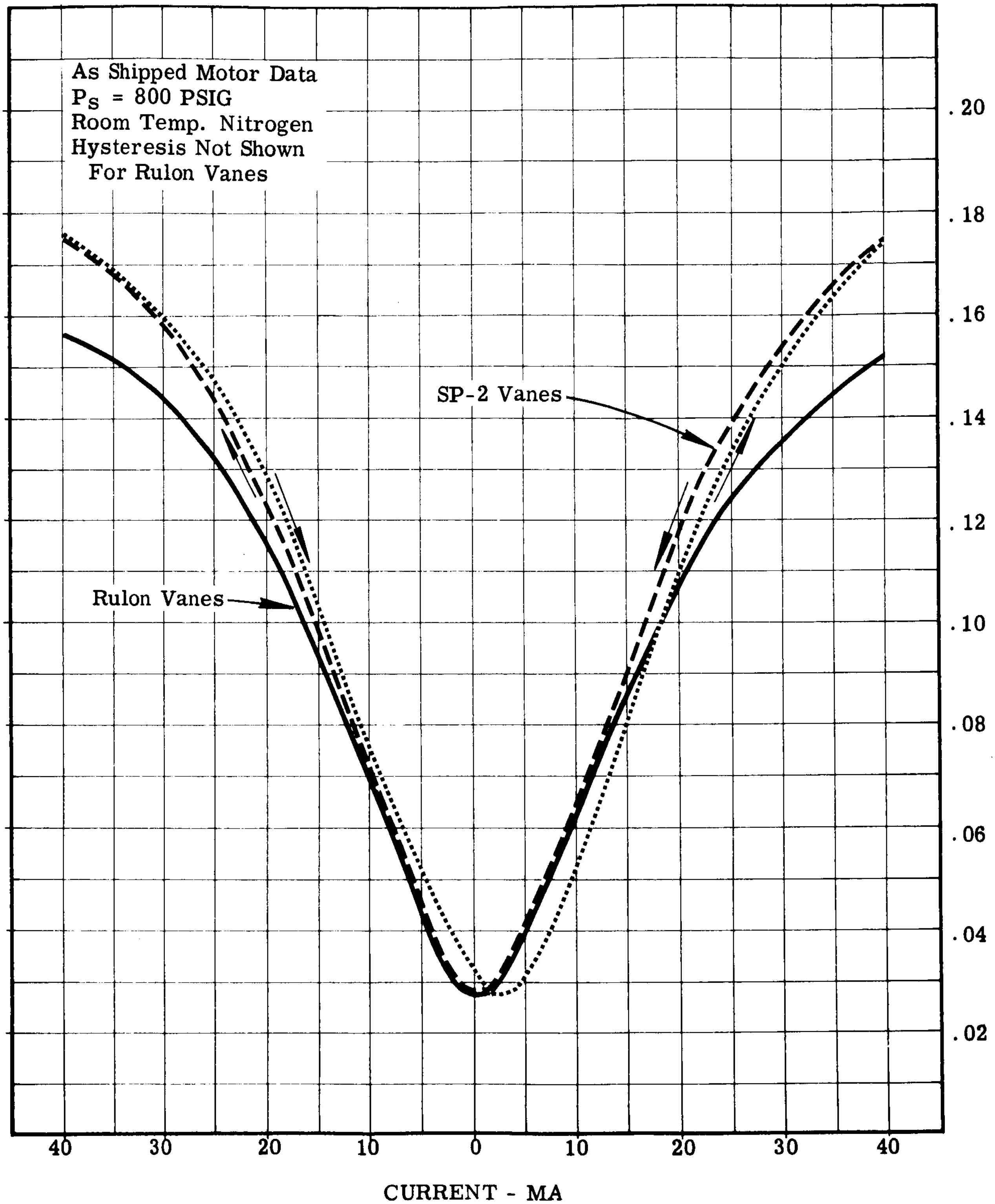


Figure 4-38. Stalled Flow Characteristics - Actuator S/N 1 Motor.

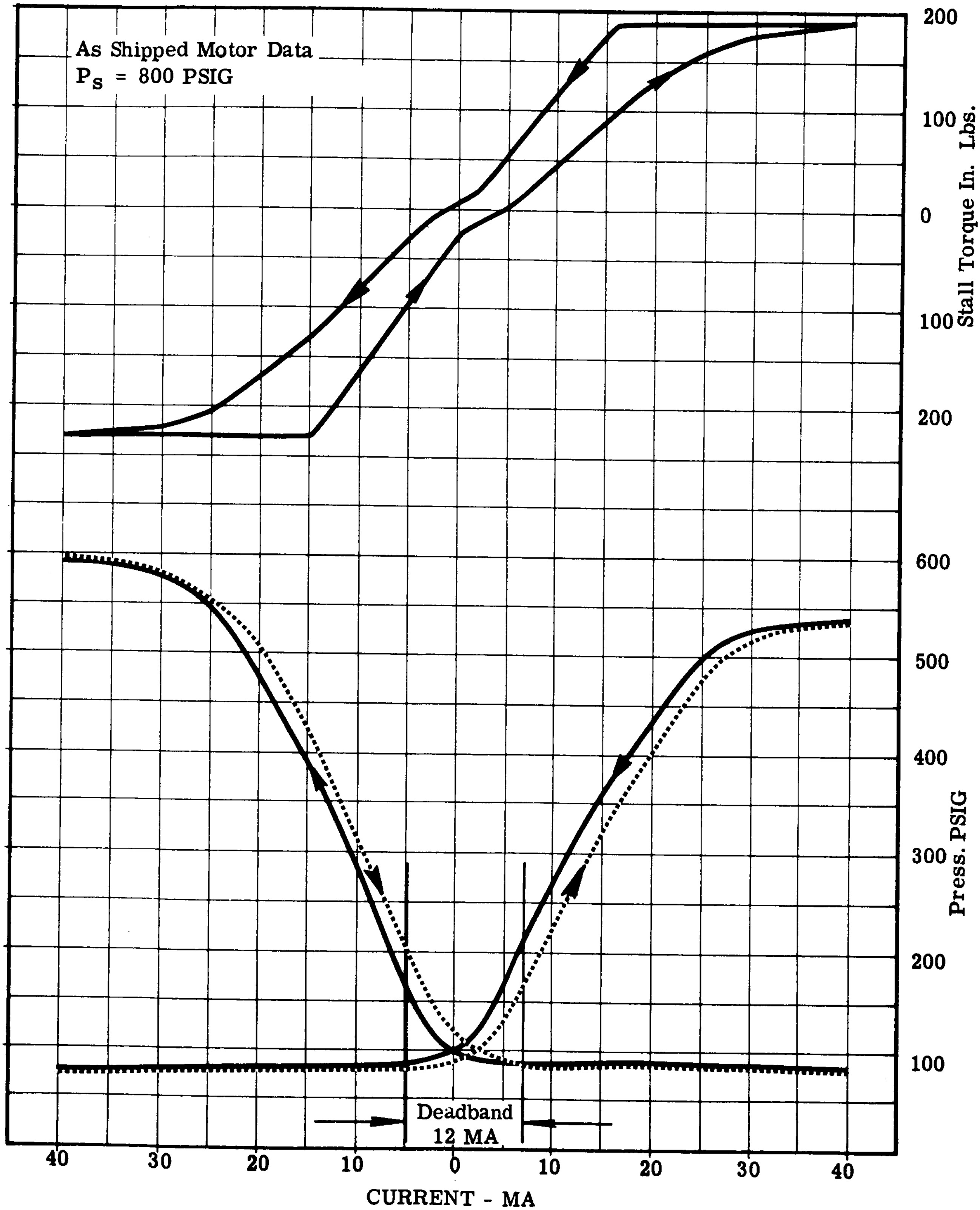


Figure 4-39. Stall Characteristics - Actuator S/N 2 Motor.

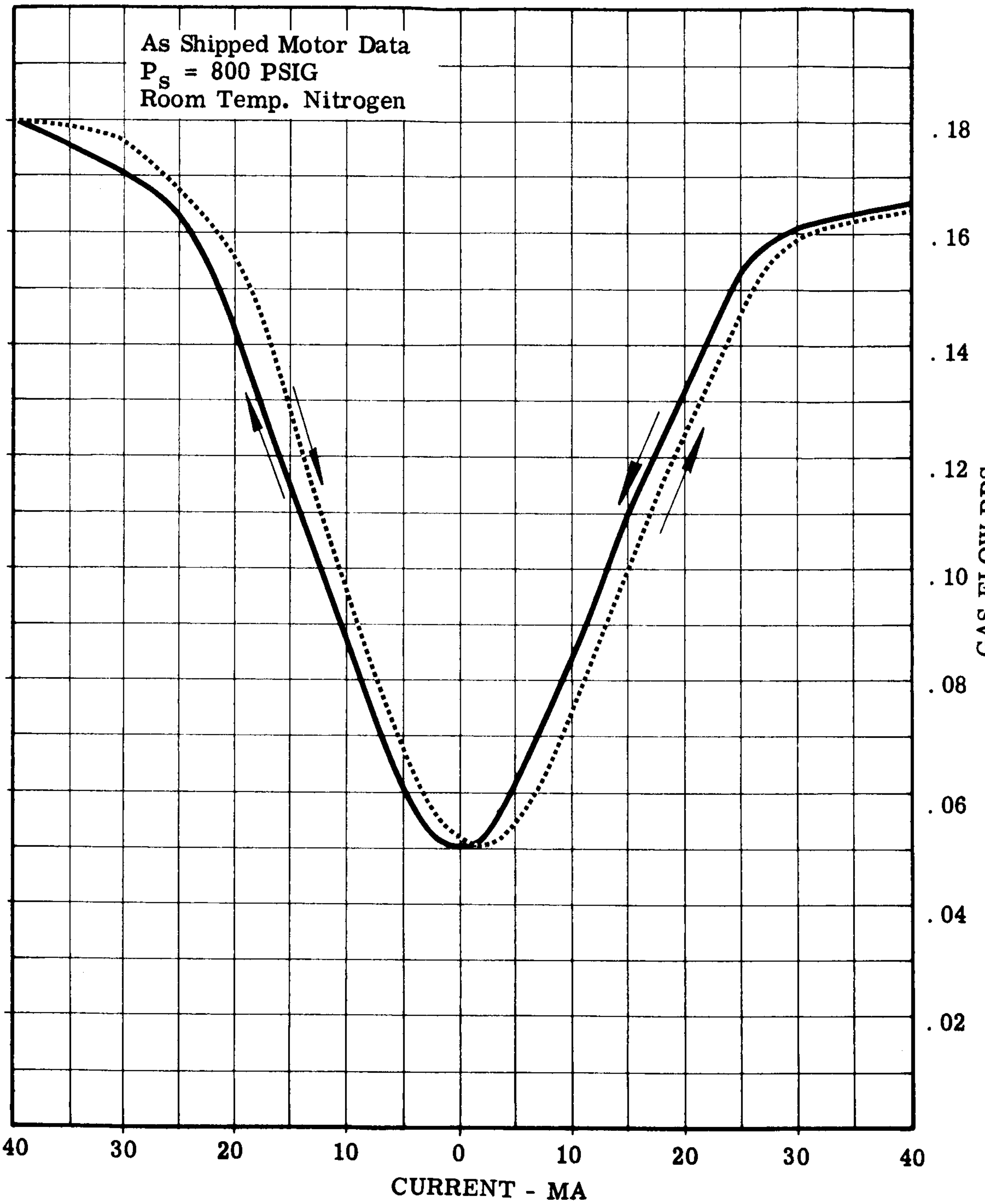


Figure 4-40. Stalled Flow Characteristics - Actuator S/N 2 Motor.

It should be noted that the torque motor used on S/N 2 motor had a greater hysteresis and the travel was not symmetrical as finally tested which accounts partially for the lower ΔP pressures obtained.

The S/N 2 servo valve used on S/N 2 motor was underlapped approximately 0.002 inch, whereas a slight overlap was used on S/N 1 servo valve on S/N 3 motor. This would account for the higher quiescent flow on S/N 2 motor.

It was found that the deadband could be reduced to as low as 6ma by using teflon end plates instead of Rulon. This permitted running in with practically no end clearance on the rotor, thereby reducing cross-port leakage to a minimum. This gave a P_1, P_2 curve as shown in Figure 4-41. It should be noted that the cross-over of P_1 and P_2 pressures is much more abrupt, the quiescent pressure is higher and, even though considerable hysteresis is shown, the pressure gain is sufficiently high to provide motor reversal in a total of 6ma. The teflon end plates did not have sufficient wear resistance to maintain this condition for more than thirty minutes of cycling.

Difficulties were experienced with the breadboard and both prototype motors with scoring of the original chrome plated end caps and ends of the aluminum hardcoated rotors. It was found that cementing 0.015 inch thick Rulon Type LD material would prevent scoring and reduce rotor friction. Both prototype motors were reworked before delivery. Inspection prior to delivery indicated very minor wear of the Rulon faces.

Rulon vanes were not available for test until just prior to delivery of the first actuator. As can be seen by comparing Figures 4-37 and 4-39, the Rulon vanes indicate less stall torque hysteresis and do not indicate a deadband increase after a period of inactivity. Rulon vanes were used in S/N 2 motor which was shipped on S/N 2 actuator.

4.2.4.4 Conclusions

With exception of the large threshold (deadband), the motor operating characteristics are satisfactory for use in the complete actuator system. The stall and rated torques, Figures 4-37, 4-39, and 4-31, are slightly lower than desired values defined in the design analysis section 3.2, but will permit evaluation of the complete actuator system.

The maximum no-load motor speed, Figure 4-32, which was run on nitrogen gas will run on hydrogen gas at -250° F exceeding the 6950 rpm desired in the design analysis section 3.2. This excessive speed will not interfere with evaluation of the actuator. The maximum speed of the motor can be corrected with a speed limiting control at a later date.

The higher threshold characteristics shown in Figure 4-39 were reduced to 6.0ma using teflon end plates. Although the teflon end plates did not have sufficient life for incorporation in the delivered units, there is sufficient evidence to anticipate a significant reduction in the threshold with additional development effort.

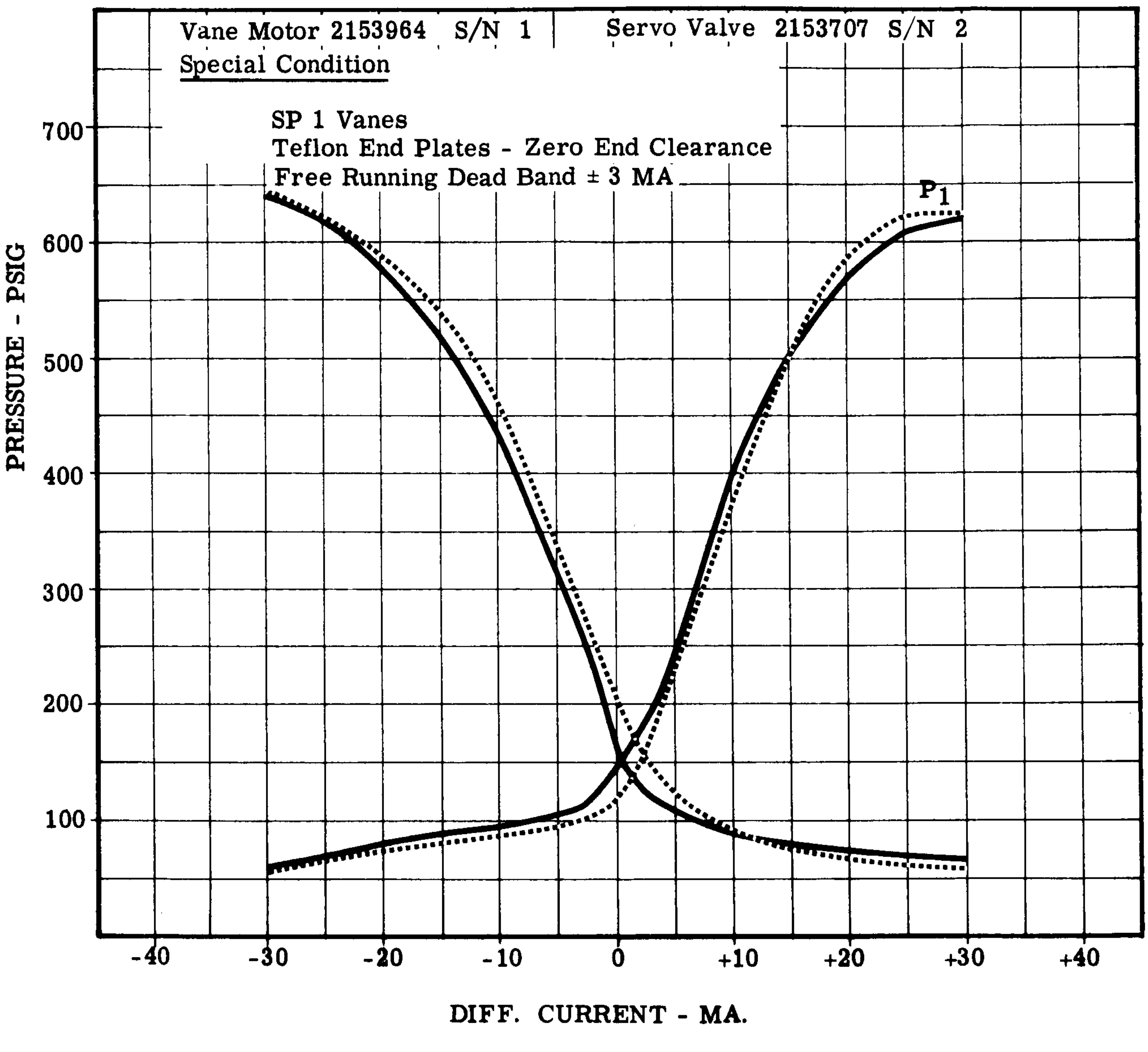


Figure 4-41. P_1 And P_2 vs. Δi (Stalled Motor).

The life of the vane motor using Rulon vanes was not demonstrated as they were introduced late in the development effort. The Rulon vanes made a significant improvement in the overall actuator performance. It is anticipated that the specified life requirements may be achieved with the Rulon vanes.

4.2.4.5 Recommendations

As reported at a technical liaison meeting at MSFC early in the development of the actuator system, the major problem area associated with the vane motor is its threshold characteristics. It is recommended that the following additional development efforts be initiated to improve the threshold characteristics and other performance parameters of the actuator system:

1. Evaluate vane configurations and vane, vane slot, and end plate materials to reduce the effects of friction on the existing vane motor design.

2. Evaluate overall system compensation techniques to reduce the effects of motor threshold.

3. Evaluate speed limiting techniques for existing vane motor design.

4. Initiate the development of a new motor concept with inherent low resolution characteristics. New motor concepts are now being investigated to utilize the lower damping, low fuel consumption, and design simplicity of the vane motor with the resolution characteristics of a gear motor.

4.2.5 Ball Screw Development

The purpose of this development effort was to determine the life of ball screws using various methods of lubrication at cryogenic temperatures. This data can then be used to provide life factors for the chosen type of lubrication for future design efforts.

4.2.5.1 Description of Parts Tested

Each of the four test assemblies used consisted of two ball nuts and a ball screw which was mounted in a test fixture for cycling under load to determine life. The ball screws tested and those used in the actuator design are compared in the following table:

Item	Test Assembly	Actuator Design
Material	440C Stainless Steel R_c 57-60	Latrobe BG-42
Screw Diameter	1.00	3.00
Ball Diameter	0.125	.2812
Ball Material	440C & Titanium Carbide	Titanium Carbide
Circuits/Nut	2 (Each Circuit 72 Balls)	3 (Each Circuit 105 Balls)
Loaded Balls/Nut	125	250

4.2.5.2 Test Equipment

The test equipment used is shown in Figure 4-42 in which the test assembly, Figure 4-43, was installed. This fixture was driven from a hydraulic power source.

Cold tests were performed by enclosing the test assembly section in an insulated chamber containing a pan of liquid nitrogen which was replenished at regular intervals. Gaseous nitrogen was then passed through a copper coil located in a dewar of liquid nitrogen to agitate the liquid in the test chamber. Thermocouples were attached to the ball nuts and an ambient thermocouple was included in the test chamber.

4.2.5.3 Test Procedure

The endurance tests were run by loading the load end nut with a pneumatic power cylinder which, in turn, loads the drive end nut by transfer of force through the ball screw. The ball screw section was then rotated sinusoidally through a total amplitude of 3 revolutions at 1/2cps. This type of arrangement loads both nuts and, since no displacement is involved, the drive motor only is required to produce sufficient torque to overcome friction losses within the two nuts. The load, stroke, and frequency duplicate the required performance in the actuator.

The preliminary tests and the accelerated life test were run at approximately the manufacturer's recommendation of 28.8 pounds/ball or 3600 pounds total load at the cyclic rate and the approximate number of revolutions of the J2 requirement. Since life is inversely proportional to the cube of the load, the testing could be accelerated. The final endurance test was run at a load of 15 pounds/ball as a check on the results of the accelerated test.

Due to the inertia of the components involved, it was necessary to shut down the drive and determine friction torque to rotate the screw with a torque wrench at various intervals during the test. Average efficiency for the two nuts could be then calculated.

Fixtures were also used to mount each nut and then rotate the screw against a dead weight load of 50 pounds in an effort to determine differences between the two nuts.

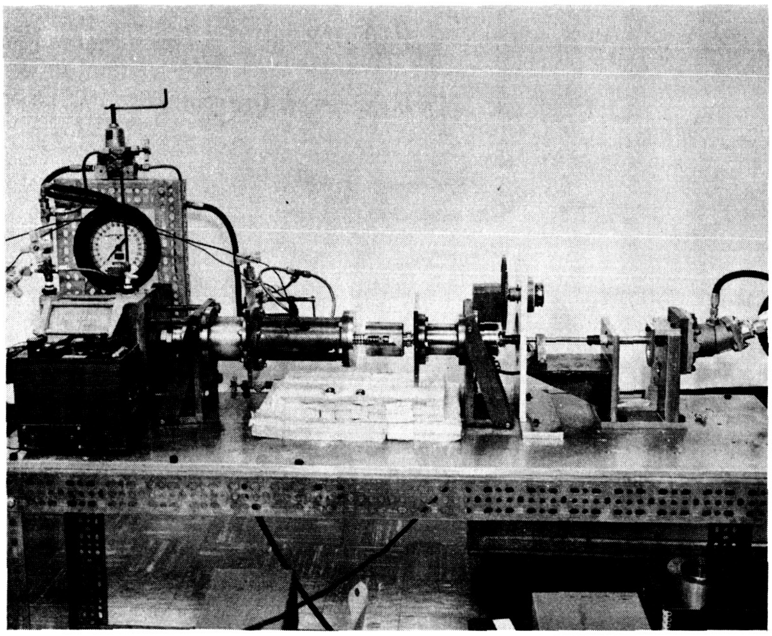


Figure 4-42. Ball Screw Test Setup.

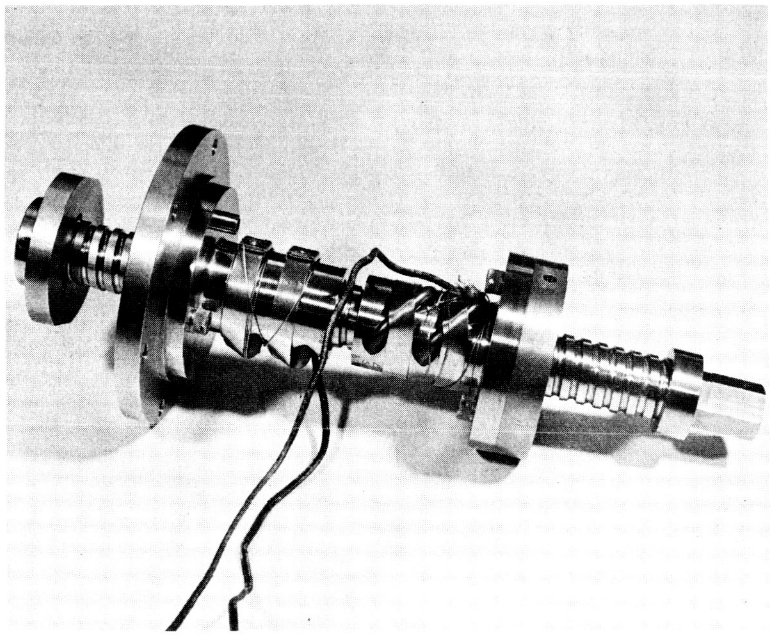


Figure 4-43. Close-up Of Ball Screw Test Assembly.

4.2.5.4 Summary

Prior to this test, several preliminary tests were run to determine effectiveness of various methods of lubrication. The use of one teflon ball out of twelve was determined to be the best method of providing lubrication of the various methods tried. General data obtained during preliminary testing is included in Section 4.5.2.5 and Table 4-3.

Two separate endurance tests were run to determine the life characteristics of the test ball screw assemblies using teflon ball lubrication. One test was run with a load per ball of 28.8 pounds (3600 pounds total). The 0.125 diameter balls used are rated at 28 pounds/ball for a life of 10^7 impacts with good lubrication. (An impact is defined as each time a ball passes one point on the nut.)

The accelerated test was run for seven hours and forty minutes (1.12×10^6 impacts) before fixture stall. Three hours of this run was at nut temperatures of -110° to -175°F. On disassembly, it was found that the drive end nut contained much more wear debris than the load nut. The teflon balls had picked up a considerable amount of this debris (see Figure 4-44). The drive nut also showed indications of fatigue breakdown of the races (see Figure 4-45). The balls and nut on the load end were still in good condition except that the 440 balls in one circuit were blued. See Table 4-1 and Figure 4-46 for tabulated and plotted data. These same parts, except for the teflon balls, had been used for four hours and twenty minutes in a prior test. The total time indicates 1.75×10^6 impacts before material fatigue was encountered.

The second test was run at a load of 1900 pounds or 15 lbs/ball which was calculated to be the load which would provide life of 7.55×10^6 impacts or the same number of impacts as required by the J-2 specification. This test was run for fifty hours and fifty-five minutes or 7.45×10^6 impacts before the drive was stalled. Up to the failure point, the efficiency of the ball screw did not fall below 85%. The maximum efficiency was 95% at the start of this test. See Table 4-2 and Figure 4-47 for driving torque and efficiency of the ball screw during this test.

On disassembly, it was noted that the nuts and screw were in good condition. The balls in the load end nut had discolored and all but one teflon ball had been completely worn away. Minor debris had started to build up on the races under the balls. On the drive nut, one circuit of balls showed no signs of wear debris whereas in the other circuit the steel balls indicated wear and the teflon balls were worn away.

The parts were all cleaned, new teflon balls were installed in the load nut, and new balls were installed in the drive nut. The cycling was then restarted and operated for twenty-one hours and fifty minutes or 3.18×10^6 impacts at 93 to 87% efficiency before the cycle rig was stalled.

On disassembly, it was found that the races in the ball nuts indicated start of fatigue failure except for the race in the circuit of the drive end nut that was not sharing the load. Accumulated wear and teflon debris evidently caused stalling of the test fixture.

(Text continued on page 4-62)

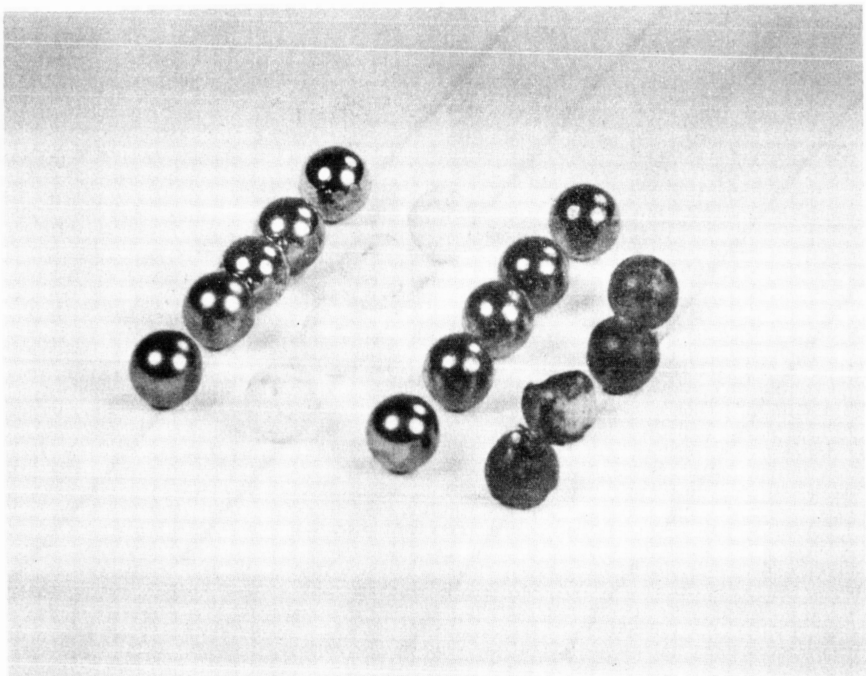


Figure 4-44. Ball Screw Test Balls.

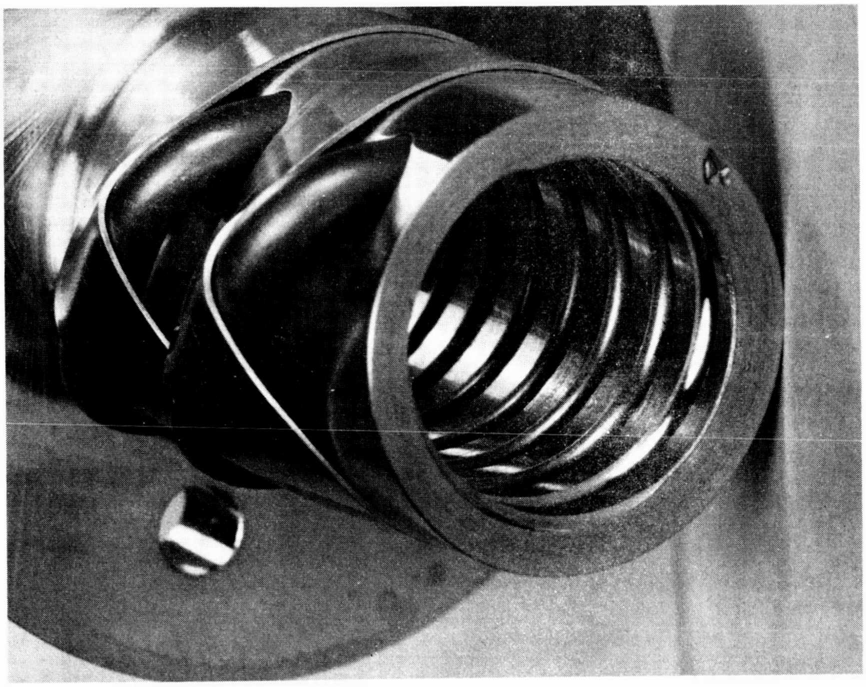


Figure 4-45. Ball Screw Test Drive Nut.

Parts Tested: Screw #5
Drive Nut #4 - Titanium Carbide Balls, Every 12th Ball Teflon
Load Nut #1 - 440C Balls, Every 12th Ball Teflon
Prior Testing At 3600 Lbs. Total Load - 4 Hours 20 Minutes

Date	Time Reading	Total	Temperature - °F Ambient	Temperature - °F Drive Nut	Temperature - °F Load Nut	Torque In. Lbs. (Max.)	Average Efficiency (Per Nut)
10-24	3:15	-	79	74	74	-	-
	3:30	15	84	150	134	-	-
10-25	1:20	No Oper.	-306	-178	-150	-	-
	1:50	Start	-292	-109	-132	25.2	91.2
	2:20	45	-325	-138	-115	23.0	92.0
	2:50	75	-320	-157	-125	23.0	92.0
	4:15	160	-322	-163	-150	25.2	91.2
	4:45	190	-318	-149	-150	25.2*	91.2
	4:50	195	-325	-213	-175	-	-
	*Note Breakaway Torque 34.5 Dropped 25.2					34.5	88.0
10-28	1:20	195	74	87	88	16.0	94.4
	1:50	225	76	151	137	16.0	94.4
	2:20	255	80	182	167	19.5	93.2
	2:50	285	81	203	194	19.5	93.2
	3:20	315	84	205	202	17.2	94.0
	3:50	345	85	205	203	19.5	93.2
10-29	9:00	345	75	118	98	26.4	90.8
	9:30	375	75	172	153	26.4	90.8
	10:00	405	75	206	193	24.2	91.6
	10:30	435	76	260	216	30.0	89.6
	10:55	460	80	289	235	79.0	72.4
	Stopped After Cooling					63.0	78.0

Static Load Test (50#) - Drive Nut 45-50 in. oz.
Load Nut 45-48 in. oz.
At Start of Test - Drive Nut 36 in.oz.
Load Nut 38-40 in. oz.

Condition of Parts After Test
Drive Nut - Start of fatigue failure. Contained large quantities of wear debris. Teflon balls dimpled from contact with load balls.
Load Nut - Races and balls in good condition. Some wear on teflon balls. One circuit load balls blue from temperature.

Table 4-1. Test Ball Screw Accelerated Endurance Test. Load = 2600 Lbs. (28.8 Lbs/Ball).

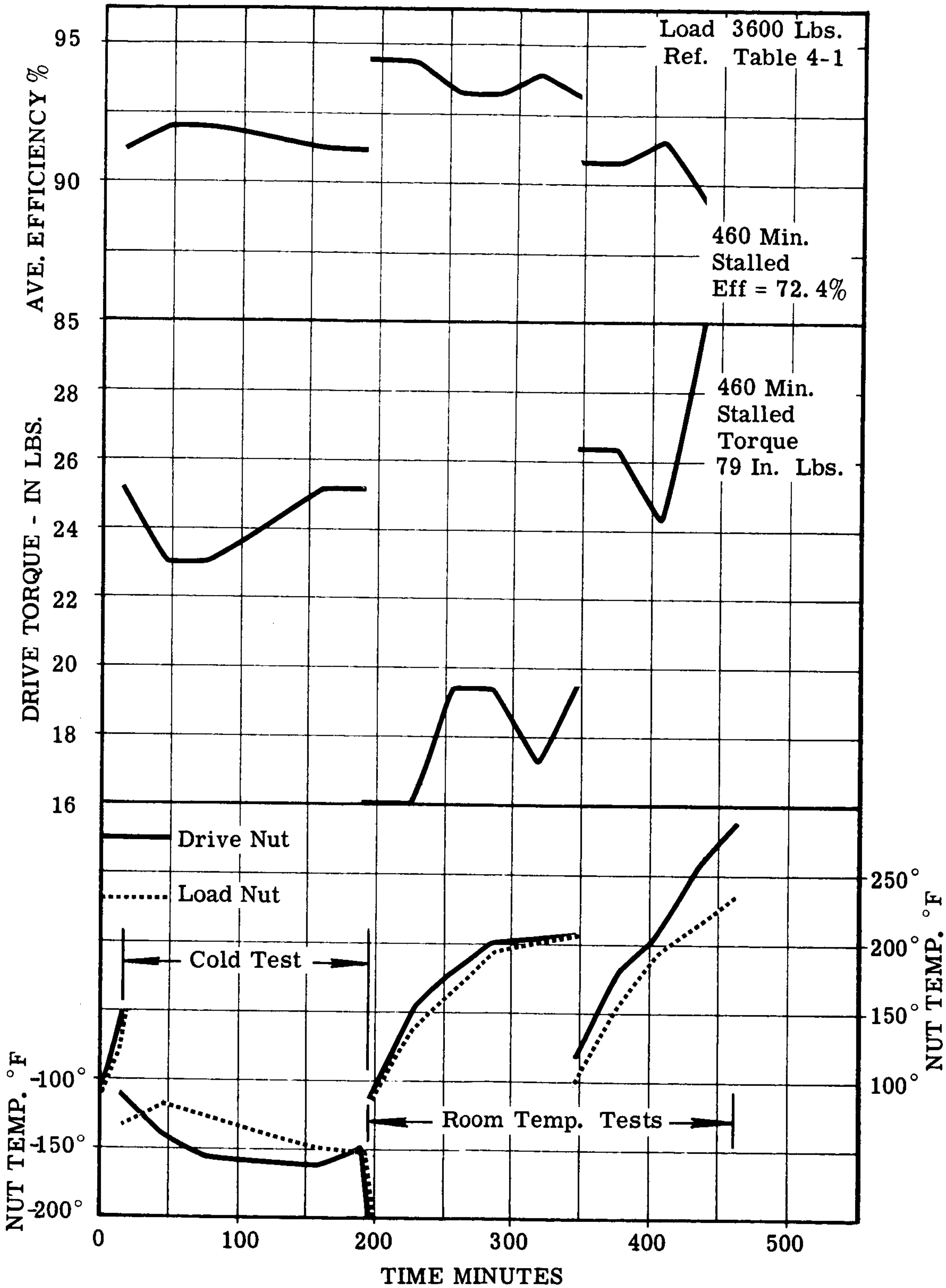


Figure 4-46. Test Ball Screw Accelerated Endurance Test.

Parts Tested: Screw #4
Drive Nut #2 - 440C Balls, Every 12th Ball Teflon
Load Nut #6 - 440C Balls, Every 12th Ball Teflon

Date	Time (Minutes) Increment	Time (Minutes) Total	Temperature - °F Drive Nut	Temperature - °F Load Nut	Torque In. Lbs. (Max.)	Average Efficiency (Per Nut)
11-20	0	0	-	-	8.1	-
	30	30	132	133	-	-
	20	50	141	138	12.7	91.7
	65	115	162	165	11.5	92.4
	60	175	170	174	14.4	90.5
	60	235	160	170	12.7	91.7
	60	295	163	174	12.7	91.7
	65	360	165	174	13.8	90.9
11-21	0	360	-	-	12.1	92.0
	30	390	140	144	15.0	90.2
	60	450	169	178	17.2	88.6
	60	510	176	191	17.2	86.6
	60	570	176	191	16.1	89.4
	60	630	179	193	16.1	89.4
	60	690	176	190	14.4	90.5
	60	750	174	190	17.2	88.6
	90	840	175	187	13.2	91.3
	30	870	175	186	11.5	92.4
11-22	0	870	-	-	14.4	90.5
	45	915	154	155	13.8	90.9
	60	975	171	183	15.0	90.2
	60	1035	182	200	14.4	90.5
	240	1275	185	197	12.7	91.7
	60	1335	183	193	11.5	92.4
	30	1365	179	188	12.7	91.7
11-26	0	1365	-	-	13.8	90.9
	35	1400	140	142	12.1	92.0
	150	1550	187	197	15.0	90.2
	90	1640	199	174	16.1	89.4
	80	1720	199	205	17.2	88.6
	100	1820	201	207	14.4	90.5
	60	1880	204	210	15.0	90.2
11-27	0	1880	-	-	14.4	90.5
	90	1970	191	197	-	-
	75	2045	200	215	16.7	89.0
	105	2150	207	223	15.5	89.8
	60	2210	209	214	15.0	90.2
	60	2270	213	221	16.1	89.4
	120	2390	202	214	17.2	88.7

Table 4-2. Test Ball Screw Accelerated Endurance Test. Load = 1900 Lbs. (28.8 Lbs/Ball).

Table 4-2 (Continued)

Date	Time (Minutes)		Temperature - °F		Torque In. Lbs. (Max.)	Average Efficiency (Per Nut)
	Increment	Total	Drive Nut	Load Nut		
11-29	0	2390	-	-	16.7	89.0
	45	2435	172	186	-	-
	45	2480	195	219	19.5	87.2
	90	2570	215	231	18.4	87.9
	90	2660	210	217	17.8	88.3
	90	2750	229	241	21.8	85.7
	90	2840	222	225	21.8	85.7
12-2	0	2840	-	-	20.7	86.4
	35	2875	191	185	-	-
	75	2950	199	209	18.4	87.9
	90	3040	209	214	15.0	90.2
	15	3055	188	210	71.0	53.0

Individual Nut Check - 50#

	Drive Nut	Load Nut
Start: 0 Time	38-40 in. oz.	37-40 in. oz.
Complete: 3055 Min.	65-90 in. oz.	80-96 in. oz.

Teardown and Inspect:

Drive Nut - One circuit 400 Balls. All well worn. Teflon balls completely worn out. Nut in good condition after cleaning. Other circuit, it appears as though the balls were not loaded.

Load Nut - Load balls in good condition. Teflon balls have all but one ball completely worn away. Nut in good condition. Screw in good condition.

Cleaned all parts. Replaced teflon balls in load nut and all balls in drive nut. Reinstalled on fixture for continued test.

Table 4-2 (Continued)

Date	Time (Minutes)		Temperature - °F		Torque In. Lbs. (Max.)	Average Efficiency (Per Nut)
	Increment	Total	Drive Nut	Load Nut		
12-6	0	0	-	-	10.3	93.0
	35	35	129	152	-	-
	15	50	135	157	10.3	93.0
	60	110	146	170	12.7	91.7
	90	200	159	195	15.0	90.1

Table 4-2. Test Ball Screw Accelerated Endurance Test. Load = 1900 Lbs. (28.8 Lbs/Ball).

Table 4-2 (Continued)

Date	Time (Minutes)		Temperature - °F		Torque In. Lbs. (Max.)	Average Efficiency (Per Nut)
	Increment	Total	Drive Nut	Load Nut		
12-9	0	200	-	-	16.1	89.3
	60	260	154	196	18.4	87.7
	60	320	168	208	-	-
	90	410	186	234	-	-
	60	470	193	242	19.5	87.2
	120	590	196	232	17.2	88.7
	45	635	200	236	19.5	87.2
12-10	0	635	-	-	17.8	88.3
	40	675	159	200	-	-
	90	765	203	253	19.5	87.2
	240	1005	201	241	-	-
	60	1065	184	203	12.7	91.7
	90	1155	164	180	12.7	91.7
12-11	0	1155	-	-	16.1	89.4
	60	1215	162	176	-	-
	60	1275	169	200	19.5	87.2
	35	1310	177	208	78.0	49.0 Stalled

Individual Nut Check - 50# Load

		Drive Nut	Load Nut
Start:	0 Time	40 in. oz.	40 in. oz.
Complete:	1310 Min.	53 in. oz.	96 in. oz.

Teardown and Inspect:

Races in both nuts showed signs of metal fatigue failure. The load nut #6 was far worse and the balls had also failed. The teflon balls of the load nut were in good condition except for large amounts of wear debris.

The drive nut balls of one circuit were still bright; other circuit blued. Teflon balls were all in good condition, very little debris.

Table 4-2. Test Ball Screw Accelerated Endurance Test. Load = 1900 Lbs. (28.8 Lbs/Ball).

It should be noted that only three hours of test on the accelerated endurance test were run at cryogenic temperatures of -175° to -95°F. It is Bendix' experience, based on past test work on the NERVA program, that, except for possible effects of brittle materials, wear under room temperature conditions is much more severe than at cryogenic temperatures. The three hours of operation at nut temperatures

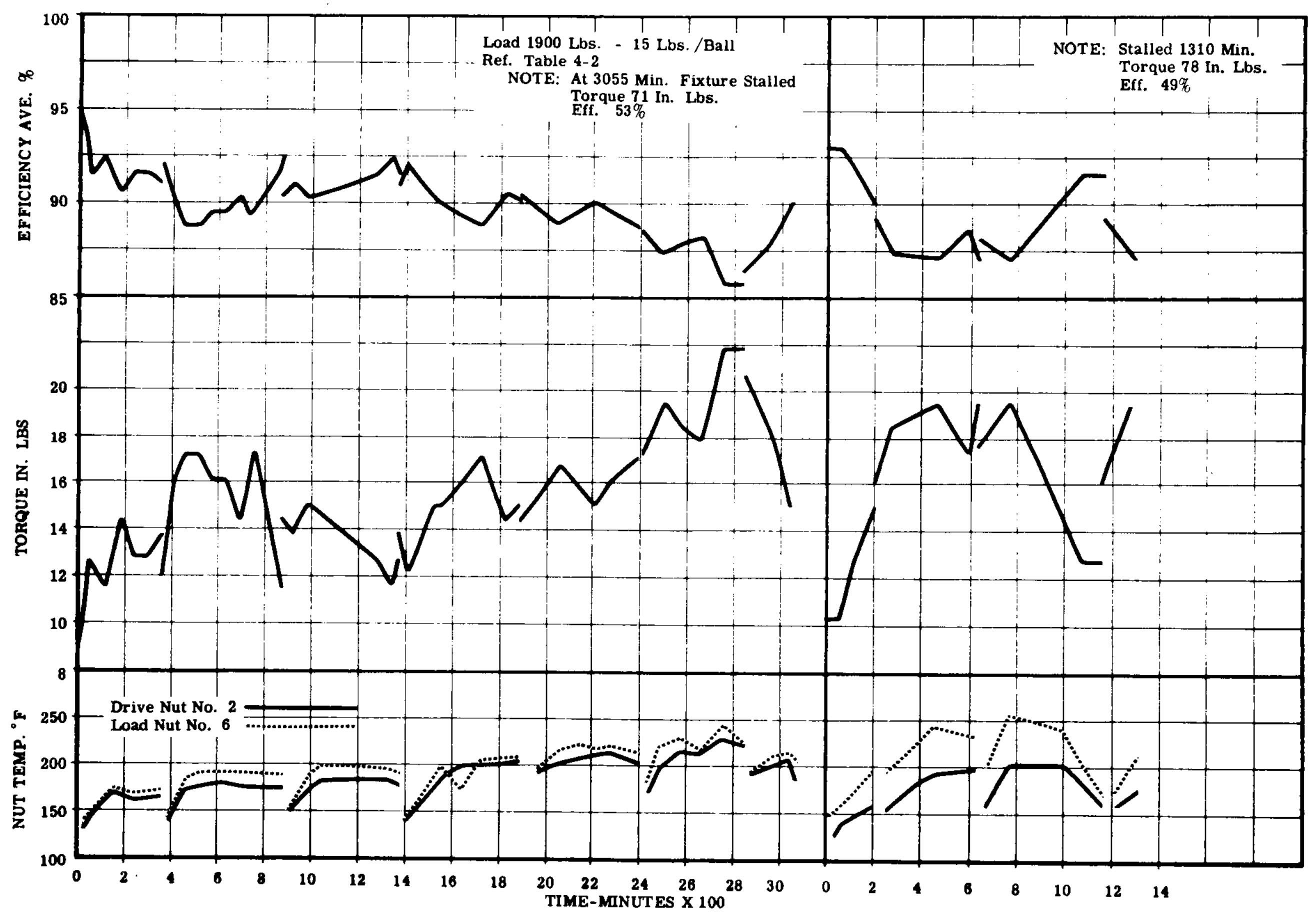


Figure 4-47. Test Ball Screw Final Endurance Test.

of -110° to -175°F, which is in the range of minimum impact strength for 440C material, was more than sufficient to check any effect of brittleness.

The titanium carbide balls used in the accelerated test indicated that more metallic wear debris was formed than with the 440C balls.

The lighter loads of the second endurance test appeared to cause less metallic wear debris. The greater number of cycles completely wore out the teflon balls before sufficient debris was accumulated to stall the test fixture.

4.2.5.5 Discussion of Other Methods of Lubrication

Prior to and between the two endurance tests reported, several other methods of lubrication were tried. These methods, along with general life characteristics, are listed in Table 4-3. Following are observations based on these lubrication tests.

Dixon Corporation's Flurocarbon Grease 95-1 was tested by applying a light coating on the screw. A run-in period of forty minutes at 1800 pounds load was used in an effort to create a film of Flurocarbon on the parts. The excess grease was then removed and a cycle test at 3600 pounds load was run. Three nuts lubricated in this manner were tested. Initial stalls were encountered in sixty-six minutes on nut #9 on the drive end, and ninety minutes for nut #9 on the drive end,and nut #0 on the load end on the second test. The first test was continued by increasing pressure on the drive motor for 192 minutes. On disassembly, the lubricant film had been worn through (probably at the sixty-six minute stall point) and sufficient wear debris had accumulated and built up on the parts to cause stall. The second test was continued to 132 minutes and, on disassembly, it was found that the lubricant film was worn through on both nuts, but only nut #0 on the load end had enough wear debris to cause stall.

Du Pont's SP-1 balls were used interspersed between load balls - one out of 6 and one out of 12. These balls were oversize by .0003 and could only be run for four minutes at 3600 pounds before stalling the test fixture. Several tries were made to wear in the balls and a total of 79 minutes was run at 1800 pounds. Examination of the parts indicated that SP-1 material would build up on the balls and races and cause stall. Du Pont has advised that in the presence of oxygen, SP-1 material has poor and erratic friction coefficients, but does not exhibit these tendencies in nitrogen or hydrogen atmosphere. The oversize balls and toughness of the material was very detrimental to these tests.

Two of Bel-Ray Company's lubricants were tested. These lubricants both deposited MoS_2 on parts. The coatings could not be made thin enough to prevent a build-up of MoS_2 on the balls and races which would stall within twenty minutes of operation at 1800 pounds load. No operation could be accomplished at 3600 pounds. MoS_2 used on the hard, fine finished surfaces of the ball screw parts would not embed in the material itself as such lubricants are intended to do.

Rulon CHR Spray (Aerosol Cans) manufactured by the Connecticut Hard Rubber Company was sprayed on the screw and run in for forty minutes at 1800

Lubrication	Parts Used			Time At 3600 Lbs.	Comments
	Screw	Drive Nut	Load Nut		
None	3		0	120 Min.	High wear and accumulated debris.
Flurocarbon Grease 95-1 (Dixon Corp.) (40 min. break-in at 1800 lbs., then excess removed)	3	9		66 Min.	Continued to 192 minutes. Teflon grease gone. Large amounts of wear debris.
	3	9	0	90 Min.	Stalled. Drive nut showed considerable wear. Debris in load nut. Total time 132 min.
SP-1 Balls. Balls were .0003 oversize. First 56 min. one (1) in 6 balls SP-1, last 25 min. one (1) in 12 balls.	4	7		4 Min.	Stalled. Continued to run total 79 min. at 1800 lbs. to wear in SP-1 balls. Build-up SP-1 on balls and races.
Molylube N (Bel-Ray Co., Inc.) Molylube SR (Bel-Ray Co., Inc.)	4 4	 7	3	20 Min.- (1800 lbs.)	Would not run above 1800 lbs. load. MoS_2 built up on balls and races.
Rulon Spray - 40 min. run-in 1800 lbs. (Conn. Hard Rubber)	3	9	0	38 Min.	Stalled. Cleaned parts, run 13 min. Balls blued nut #9.
Teflon Balls One (1) out of 6	3		0	66 Min.	Stalled. Continued total 180 min. Two 440C balls had failed.
Teflon Balls One (1) teflon out of 12 Light 95-1 grease on screw	3	9	0	35 Min. 26 Min. 17 Min.	Stalled) Excess teflon apparently Stalled) caused screw to stall. Stalled)
Rulon Balls Nut #6, 1 out of 12 Nut #2, 1 out of 6 Balls .002 oversize	6	6	2	110 Min.	Stalled. Cooled - ran for 30 min. and stalled. Rulon built up on races.

Table 4-3. Preliminary Test Summary - Various Lubricants.

pounds to form a film on the parts. At 3600 pounds, the film broke down in thirty-eight minutes and the drive was stalled. Parts were cleaned, but the test failed in thirteen minutes. The balls in drive nut #9 had turned blue.

Rulon balls were also tested using one out of 12 in drive nut #6 and one out of 6 in load nut #2. These balls were oversize by .002 and the test was stalled in 110 minutes. There was some indication of Rulon build-up on the races after a total run of 140 minutes. It is felt that high thermal expansion of Rulon may have been the prime cause for stall.

Two preliminary tests using teflon balls were run prior to endurance testing. The actual results were not better than that of other lubricants tested. When one teflon out of 6 balls were used, the test fixture stalled in sixty-six minutes. Disassembly after 180 minutes indicated that two 440C balls had failed - the probable cause was balls sliding rather than rolling. In the other test, a very light coating of 95-1 Flurocarbon Grease was used on the screw along with one teflon ball out of 12 balls. This combination resulted in excessive amounts of material which stalled in thirty-five minutes.

It was noted, however, that the teflon balls did form a very light film of teflon on balls and races. This film did not build up as did SP-1, MoS_2, and Rulon. Excessive teflon deposits could cause trouble if too many teflon balls or additional teflon were used. The teflon balls tended to pick up minute particles of metallic wear debris.

It should be noted that, without lubrication, a nut was operated for 120 minutes before accumulated wear debris caused the fixture to stall. This indicates that other dry film-type lubricants, except teflon supplied by the use of one teflon ball out of 12, actually caused stall in shorter periods of time than running the parts without lubrication.

4.2.5.6 Conclusions

It can be concluded from these endurance tests that the use of teflon balls interspersed approximately one out of ten or twelve load balls should provide a lubrication film that will meet the life requirements for the J-2 actuator as it is now designed.

The endurance tests indicated that one teflon ball out of twelve in a ball screw will provide sufficient lubrication for a life of 7.45×10^6 impacts with a load of 15 lbs/ball for 0.125 diameter balls. The J2 ball screw is designed for a load of 46.5 lbs/ball whereas these endurance tests indicate that the 0.281 diameter ball lubricated with teflon balls should be capable of 64 lbs/ball.

The J2 actuator has a further margin in that the Latrobe Steel BG-42 material used has a minimum hardness of R_c 60 whereas the 440C material tested has a minimum hardness of R_c 57.

Future designs using this type of lubrication should be based on a maximum load capacity of 50% of the manufacturer's rated load for well lubricated ball screw assemblies rather than the 36% used in this design.

It should be noted that prior testing using various other forms of lubrication - MoS_2, etc. - indicated that the ball screw manufacturer's recommendations should be derated to 25% to provide equivalent life. The prime difference appears to be that the teflon balls will form a film on the balls and races to provide lubrication and keep the film replenished. Teflon itself will not build up on the parts to reduce clearance; therefore, failure will not occur until such time as the teflon balls are completely worn away and teflon plus metallic wear debris starts to build up on the races or metal fatigue occurs.

4.2.5.7 Recommendations

The 50% of manufacturer's rated load capability for teflon ball lubrication, as concluded from this test, requires additional testing to verify this finding and raise the confidence level in this factor. It is therefore recommended that the J-2 actuator ball screws, as designed, be tested to failure. These ball screws differ greatly in size and materials which could affect the above factor. The proportionate number of teflon balls required may, however, need to be increased because of the larger parts.

Consideration should be given to using teflon balls in full compliment ball bearings for cryogenic use.

Rulon balls should also be considered in future test programs. These balls should be .001 to .002 smaller than the load balls.

Titanium carbide balls, used in an accelerated test, indicated greater wear of races because of the larger amount of wear debris. Comparative tests should be run between 440C and titanium carbide balls.

4.2.6 Linear Motion Potentiometer

Five potentiometers manufactured by the Markite Corporation were tested but only one unit was subjected to the vibration test.

This potentiometer is of the infinite resolution type, having dual elements and center taps.

Mechanical Stroke	3.5 ± .020 inches
Electrical Stroke	3.15 inches minimum

The following are the results of these tests:

4.2.6.1 Mechanical Stroke

Serial No.	Actual Stroke (Inches)	Serial No.	Actual Stroke (Inches)
4A70	3.485	3AZ205	3.472
3AZ204	3.497	3AZ206	3.490
		3AZ207	3.479

Potentiometer, S/N 3AZ205, stroke was .008 inch less than the minimum specified; however, the available stroke is more than sufficient for use in the NV-B1 and is therefore acceptable.

The output shaft seized solid at cryogenic temperatures. This condition was overcome by increasing the bore diameter of a teflon bushing by 0.006 inch. This provided for the differential coefficient of thermal expansion of the output shaft and the bushing.

4.2.6.2 Element Resistance

The value of the protective resistance and the element resistance is shown in the following table:

Terminals	S/N 3AZ206	S/N 3AZ204	S/N 3AZ205	S/N 3AZ207
B-D	514Ω	558Ω	538Ω	687Ω
H-F	499	544	512	503
A-C	2289	2045	2191	2069
E-G	2308	2138	2147	1983

4.2.6.3 Linearity and Electrical Stroke

4.2.6.3.1 Electrical Stroke

Each potentiometer satisfied the 3.15 inch minimum electrical stroke requirement, and was in fact equal to the mechanical stroke.

4.2.6.3.2 Linearity

The linearity was within the specified range at both room and cryogenic temperatures. Typical data is shown in Figures 4-48 and 4-49 which are room temperature and cryogenic tests respectively.

The most noticeable difference between the plots is the increase in hysteresis from .02V maximum at room temperature to .08V at -300°F.

4.2.6.4 Wiper Continuity

The potentiometers were tested for continuity at -260°F with a 25K ohm resistive load on each element. Potentiometers S/N 3AZ204 and S/N 3AZ205 gave a continuous output throughout the total stroke from each element.

S/N 4A70 gave a discontinuous output on Terminals B-D over a 2-volt travel range. The signal became continuous after 3 cycles, but would become discontinuous after a 20-minute time interval. Increasing the load from 25K to 100K ohms reduced amplitude of the discontinuity.

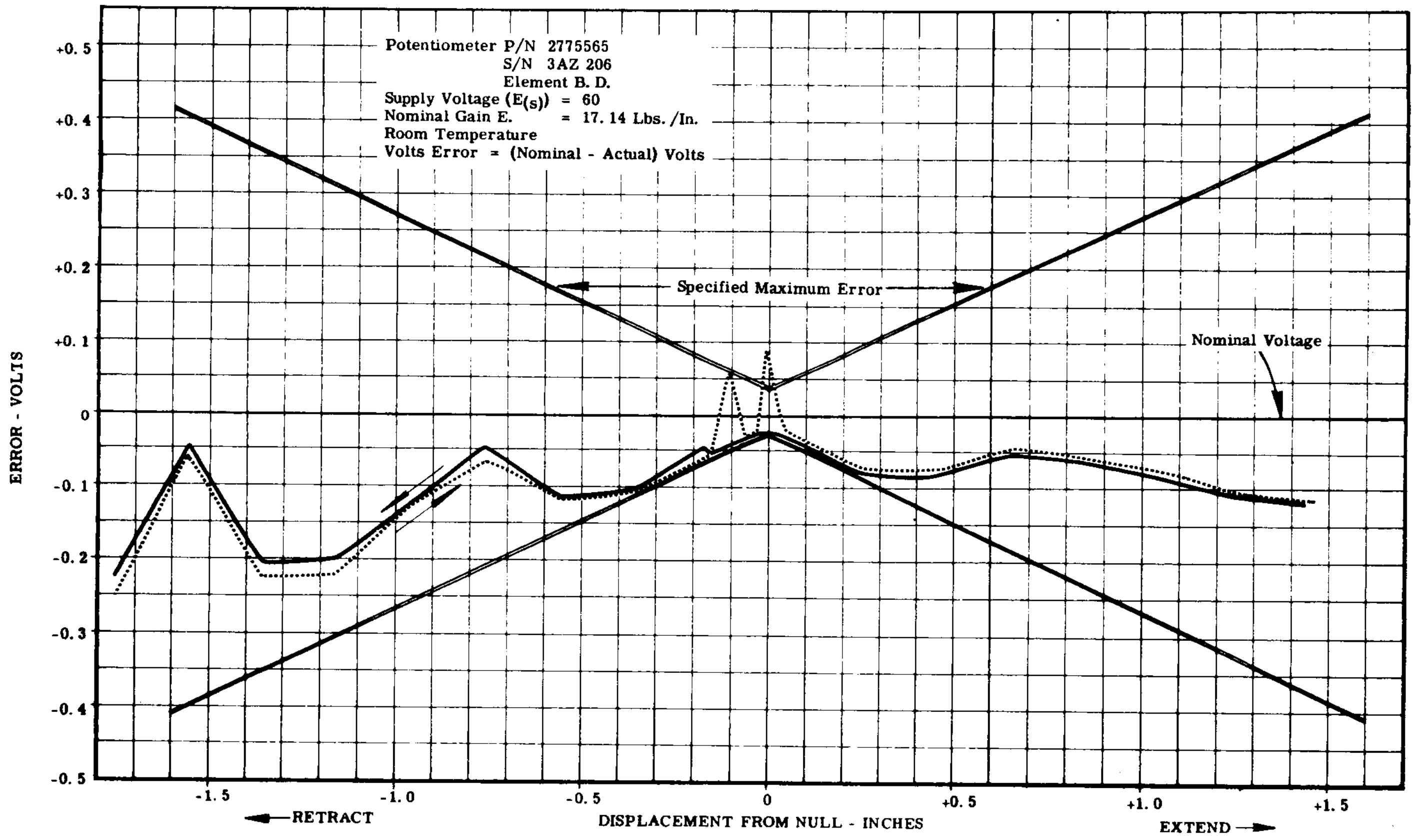


Figure 4-48. Potentiometer Error vs. Shaft Position (Room Temperature).

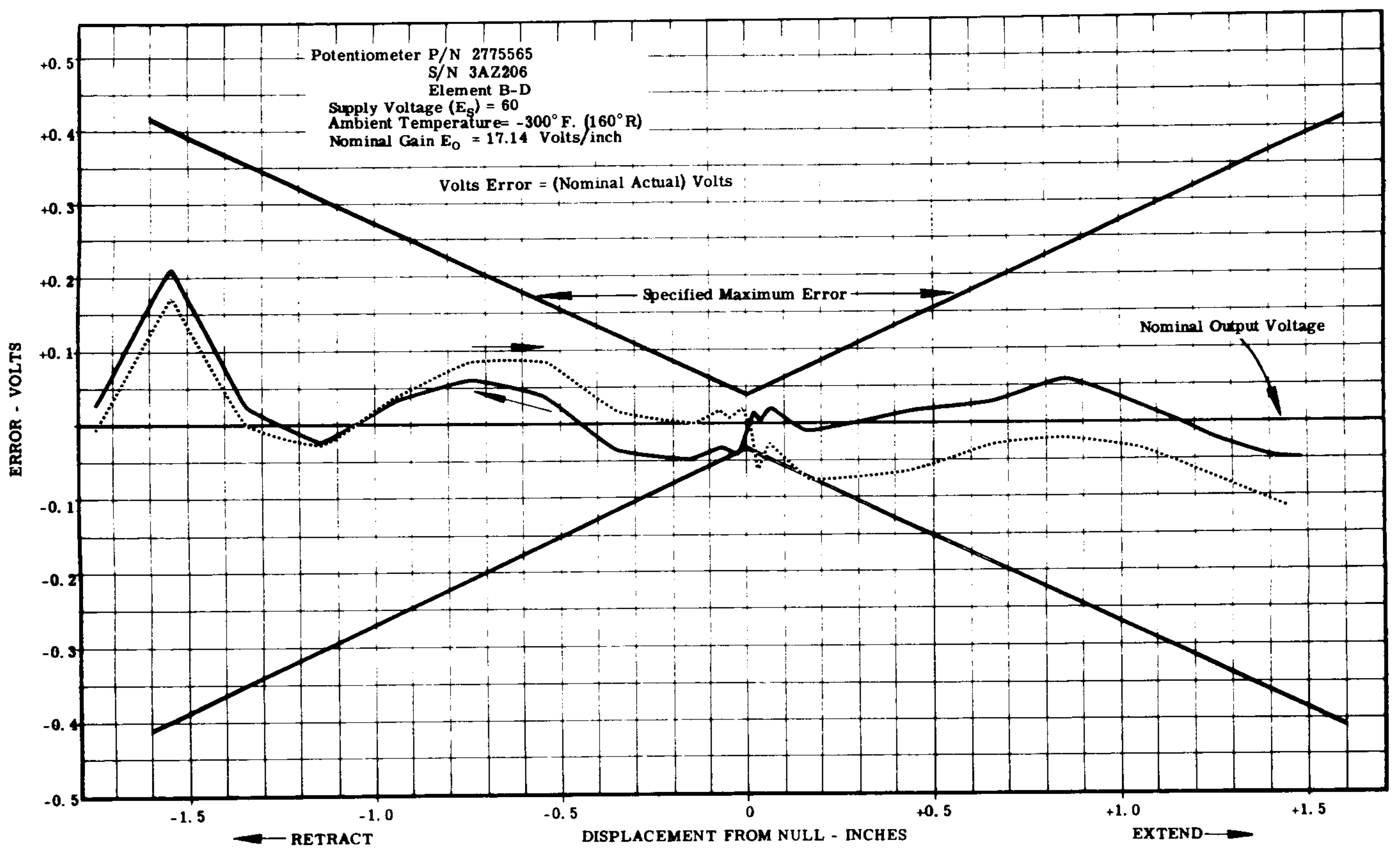


Figure 4-49. Potentiometer Error vs. Shaft Position (-300° F).

S/N 3AZ206 gave a discontinuous output on Terminals F-H at one point only, this condition was not repeatable.

S/N 3AZ207 gave a discontinuous output on Terminals F-H at three positions, this condition was repeatable.

4.2.6.5 Vibration

The potentiometer, S/N 3AZ207, failed to pass the first vibration test, both elements breaking continuity in each of three vibration axes.

The method of locating the potentiometer in the housing was modified to permit support at both ends and also to ensure full engagement of the connector.

The modified assembly was subjected to the vibration test and no discontinuities were observed.

4.2.6.6 Conclusions

The potentiometers will satisfy the initial requirements of the NV-B1 actuator but further tests are required, particularly the cryogenic wiper continuity tests before recommendations for flight hardware can be made.

4.3 DYNAMIC TEST OF ACTUATOR ASSEMBLY

4.3.1 Test Fixture

Bendix designed a walking beam engine simulator fixture in order to test the actuator under an inertia load of 1400 slugs and a friction load of 7300 pounds. This fixture is shown in Figure 4-50. As designed, a cantilever beam from a floor mounted pedestal was calculated to simulate the combined spring rate of the vehicle and engine of 391,000 lbs/in. The head end of the actuator was mounted to this spring with the rod end connected to the walking beam. Tests indicated, however, that other structural spring rates within the fixture lowered the overall effective rate to the point that resonance occurred at 4cps; with a spring rate of 391,000 lbs/in. resonance should not have occurred before 9.2cps. Design modifications were conducted which raised the spring rate of the beam support shaft. The actuator was then mounted between a hard support and the beam. A more complete description of the load fixture is given in section 4.3.2.9 titled, "Discussion Of Results."

As will be noted in Figure 4-50, a strain gage has been mounted on the rod end attaching clevis to record actuator output force. A similar strain gage is mounted on the connecting rod between the walking beam and brake arm. A linear potentiometer is attached to the right end of the walking beam to record inertia travel. The position indicator section of the actuator feedback potentiometer was used to record actuator position.

Friction load was provided by regulated hydraulic pressure to the nonservo brake.

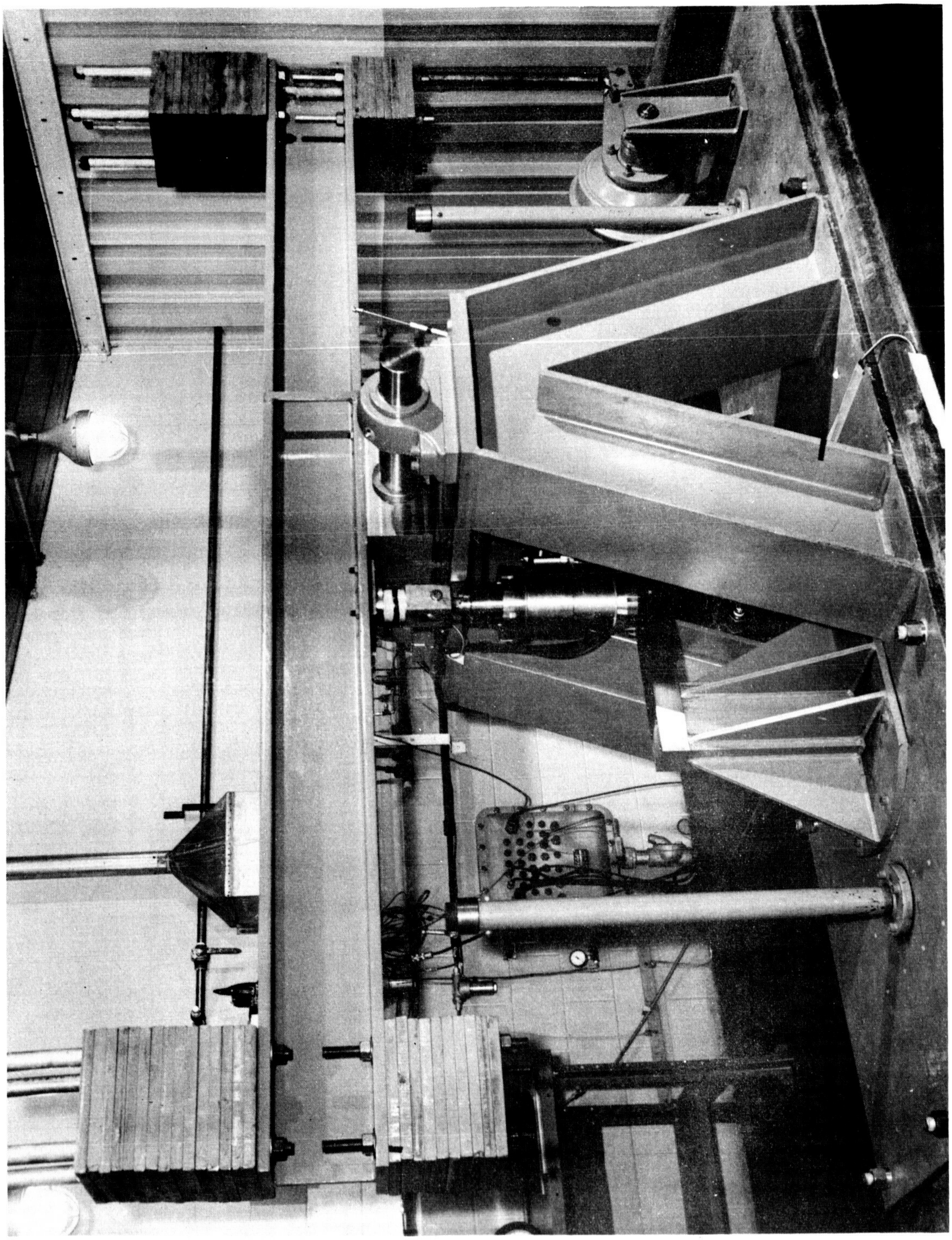


Figure 4-50. Engine Simulator Test Fixture.

A hydraulic load cylinder was solid mounted in an identical position to the actuator on the right end of the beam to provide the rated load of 23,200 pounds. This cylinder was a double shaft cylinder (equal area each end) with interconnecting plumbing so that a relief valve setting would produce the same resistive load in each direction. The cylinder could also be connected to a hydraulic supply to load the actuator in each direction.

4.3.2 Control System Open and Closed Loop Testing

The following describes the pneumatic and electrical systems, test procedures, load fixture and control system instrumentation, and presents a summary of the tests, forcing functions, test results, and the corresponding test data.

4.3.2.1 Pneumatic System

The pneumatic system schematic for both room temperature and low temperature testing is shown in Figure 4-51.

The three-way electrically controlled ball valve permits operation of the control system on either room temperature hydrogen or nitrogen by actuating a switch remotely located from the test cell. To facilitate rapid shutdown of the control system servo pressure should an emergency arise, an electrically controlled ball valve remotely actuated was placed in the servo supply line. Both hydrogen and nitrogen supply pressures were remotely adjustable.

To prevent foreign material within the gas lines from entering the actuator, two-micron filters were used in both the hydrogen and nitrogen supply lines in addition to one in the servo supply line.

Low temperature hydrogen gas was obtained by passing ambient temperature hydrogen gas through a liquid nitrogen cooled heat exchanger. The temperature of the servo gas was set by controlling the boil off temperature of the liquid nitrogen by regulating the pressure within the nitrogen chamber.

The low temperature servo supply pressure was regulated upstream of the heat exchanger where ambient temperature gas entered. The temperature of the servo gas was measured downstream of the heat exchanger. The temperature drop from the heat exchanger to the actuator servo inlet was not significant.

A five-micron filter was used upstream of the heat exchanger and a two-micron cryogenic filter was used downstream.

To measure gas flow, a Daniel sharp edged-orifice flow meter was placed in the servo supply line.

4.3.2.2 Control System Wiring

The control system wiring diagram is shown in Figure 4-52.

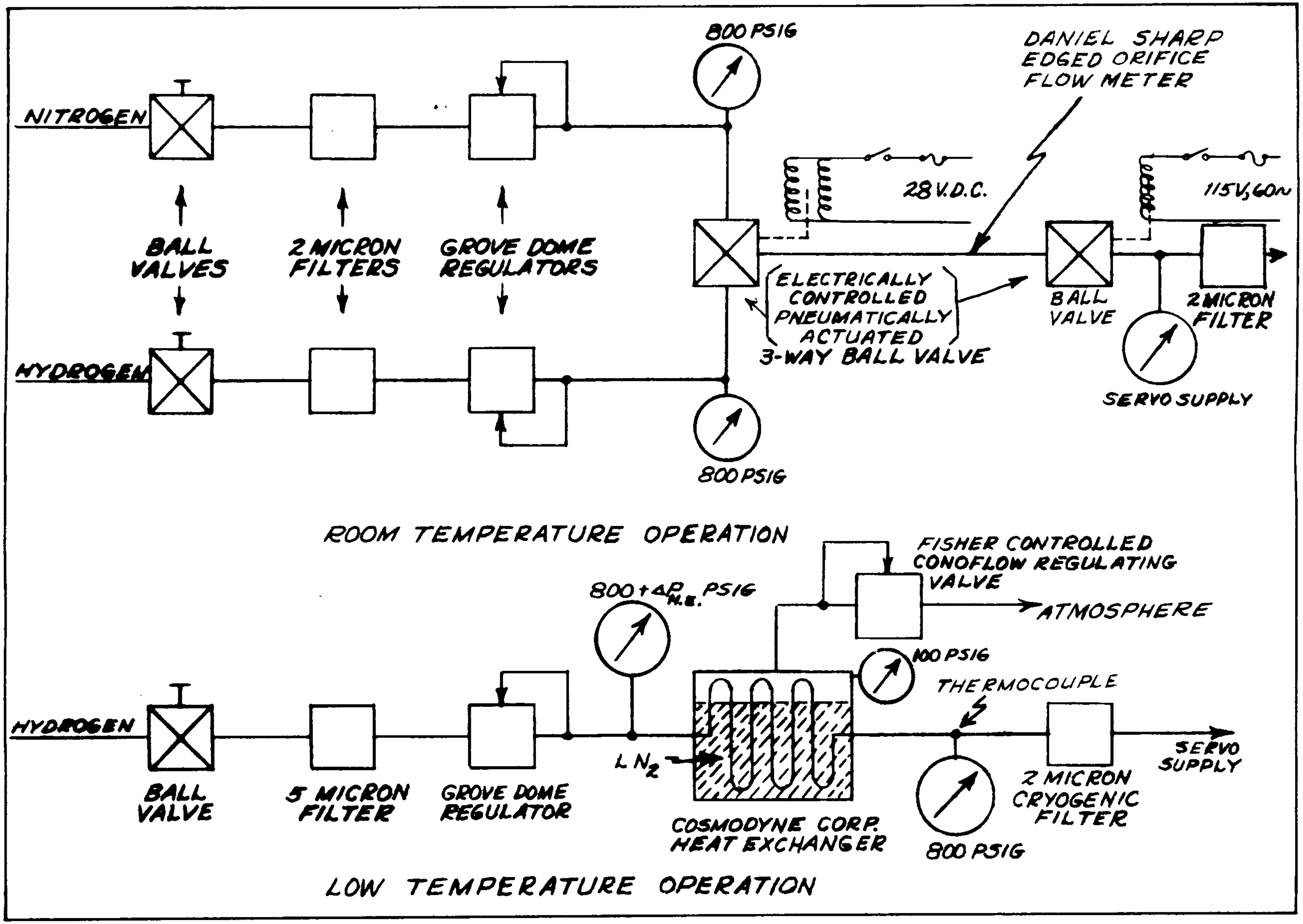


Figure 4-51. J-2 Pneumatic System Schematic.

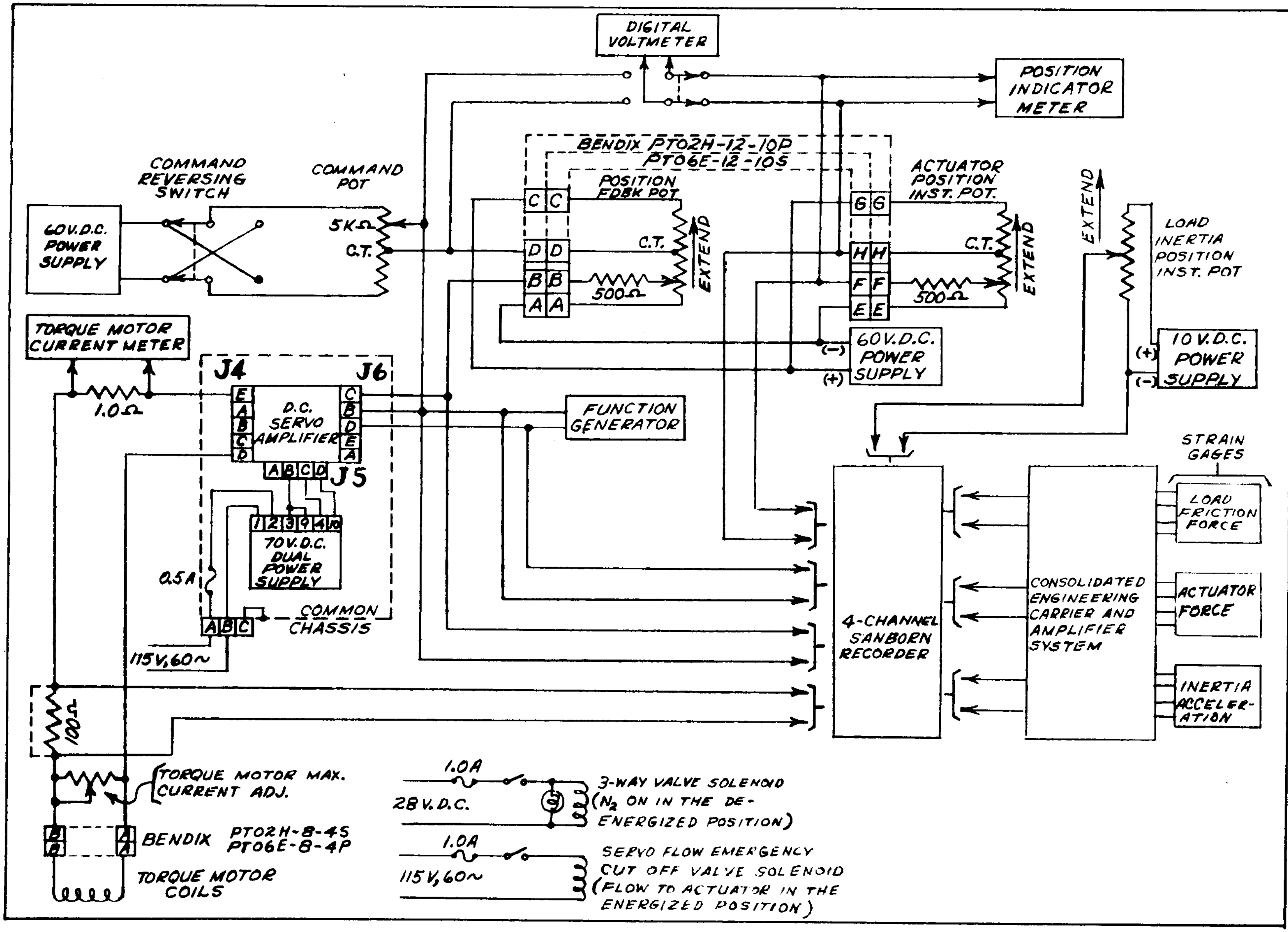


Figure 4-52. J-2 Control System And Instrumentation Wiring Diagram.

The torque motor and potentiometers are terminated at their connectors according to the specification.

The servo amplifier consists of two units, the amplifier proper and a dual 70VDC power supply, both of which are mounted on a common chassis. Supply power to the whole unit was applied through the connector mounted on the common chassis and labeled 115V, 60 cps. Connections for torque motor current, amplifier power, and input signals were made through connectors labeled J4, J5, and J6 respectively.

The amplifier has three separate ground referenced inputs available on terminals "C, D, and E" of connector J6. Terminal "B" is signal common. The actuator position error was connected to terminals "B" and "C" with the command potentiometer wiper connected to "B" or signal common. The function generator was connected to terminals "B" and "D" and the spare input terminal "E" was connected to signal common.

Amplifier output current was applied to the torque motor through terminals "D" and "E" of connector J4 and a one ohm instrumentation resistor. When load inertia acceleration and actuator velocity with a constant load as a function of torque motor current were desired, an additional one hundred ohm and decade resistor network was placed in the circuit. All connections to the torque motor circuit were ungrounded as is necessary. Terminal "D" of connector J4 is connected to terminal "A" of the torque motor connector.

The 5K ohm command potentiometer and the dual element actuator potentiometer were energized with 60VDC ungrounded regulated power supplies.

The only ground available in the system is the wiper of the command potentiometer which connects to signal common.

4.3.2.3 Load Fixture Instrumentation

The schematic of the load fixture and its associated instrumentation and their placement is shown in Figure 4-53.

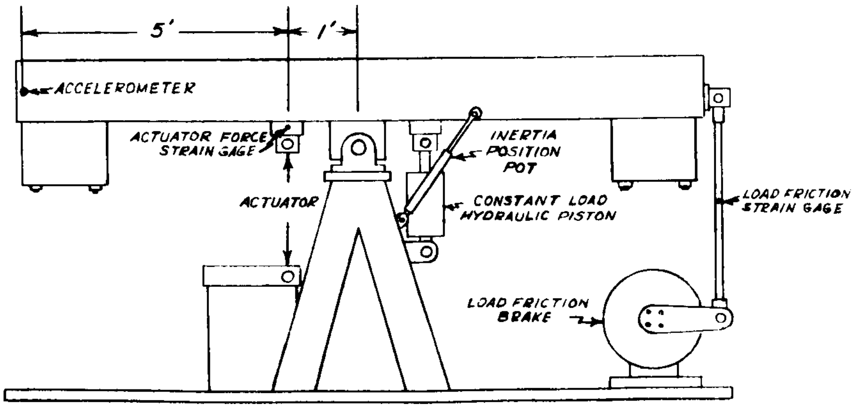


Figure 4-53. Load Fixture And Instrumentation Schematic.

The load fixture was instrumented to provide load inertia acceleration, load inertia position, load friction force, and actuator force. The accelerometer was placed at a location six times further removed from the beam pivot than the actutor attach point to improve its resolution. The load friction force was determined by measuring the elongation of its connecting rod with a strain gage and calibrating this with respect to the force generated at the actuator attach point. The load inertia position was determined by measuring the displacement of a fixed point on the beam with respect to the beam supporting structure by means of a potentiometer and calibrating this with respect to the displacement of the actuator attach point. The constant force generated by the hydraulic piston was measured at the actuator attach point with a strain gage.

The load friction force, load inertia acceleration, and actuator force signals were fed into a Consolidated Engineering carrier and amplifier system and converted to DC signals for recording purposes.

4.3.2.4 Control System Instrumentation

The control system instrumentation is shown in Figure 4-52.

The control system was instrumented to provide visual readout of torque motor current and actuator position, and for recording purposes,actuator position, actuator dynamic forcing function, actuator position error, and torque motor current. The actuator position error was provided for diagnostic purposes should the control system fail to perform as expected. The control system was also instrumented to provide steady-state command and actuator position through a D.P.D.T. switch for determining the closed loop position calibration of the control system.

Visual readout of torque motor current was obtained by measuring the voltage drop across a one ohm resistor in series with the torque motor and calibrating the meter in milliamperes. Visual readout of actuator position was obtained by measuring the voltage from the spare element of the actuator dual element potentiometer and calibrating the meter in inches.

Pertinent connections throughout the control system were made to provide voltages corresponding to actuator position error, actuator position, and actuator dynamic forcing function. For measuring the torque motor current, the voltage drop across a one hundred ohm resistor in series with a parallel combination of the torque motor and a decade resistor was measured. Since the amplifier is essentially a current source, the voltage drop across the one hundred ohm resistor will remain the same for any value of decade resistance. Thus, this voltage only indicated initiation and level of the amplifier output current,not the current supplied to the torque motor.

The current to the torque motor for a given value of decade resistance when the amplifier was saturated was determined prior to any actuator testing.

4.3.2.5 Test Procedure

For all tests, the servo supply pressure was maintained at 800 psig.

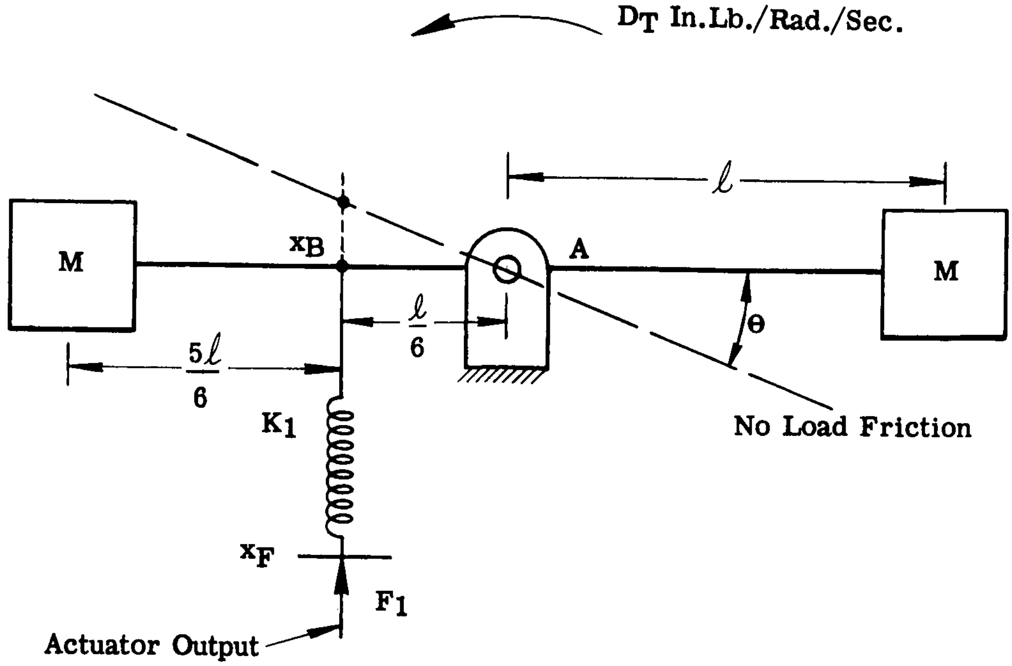


Figure 4-54. Specified Load Fixture Configuration.

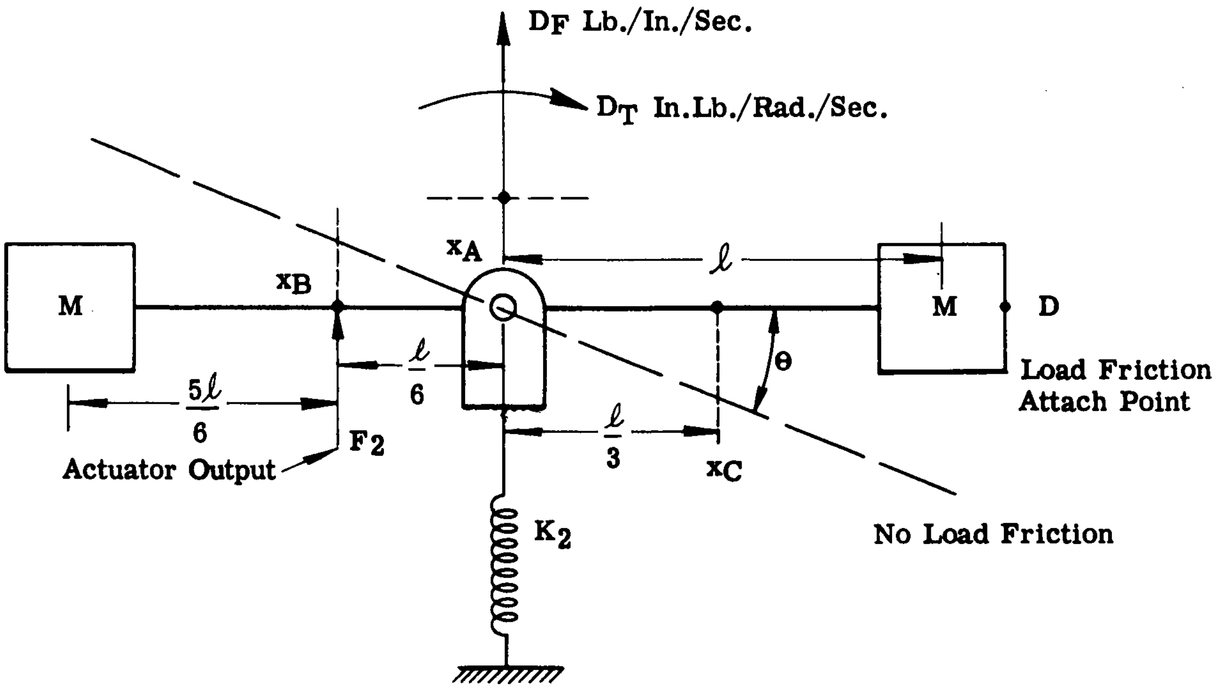


Figure 4-55. Test Load Fixture Configuration.

The command potentiometer was used to establish the nominal operating position of the actuator about which dynamic data was taken.

All dynamic forcing functions were supplied from a Hewlett Packard Model 202A function generator. The amplitude, frequency, and wave form were set according to the actuator specification.

When the actuator velocity with a constant load and the load inertia acceleration both as a function of torque motor current were desired, the maximum torque motor current was set previous to the test by adjusting the decade resistor in parallel with the torque motor. The control system was left in the closed loop configuration and a step input signal which would exceed that required to saturate the amplifier was applied at the amplifier input. The signal input was limited to less than full stroke to prevent excessive pounding of the snubbers.

For all other tests, the resistive network was removed.

To evaluate snubbing, the control system was left in the closed loop configuration and a step input signal was applied at the amplifier input. The magnitude of the step was such that the total command was larger than that required for full stroke by an amount necessary to saturate the torque motor, and of such duration as to allow maximum actuator velocity to be attained prior to snubbing.

The force generated by the constant load hydraulic piston and the load friction brake was set by applying a constant hydraulic pressure to each. The hydraulic pressure supplied to the brake was applied to the brake shoe actuating piston.

4.3.2.6 Data Recording

All dynamic data except for gas consumption were recorded on a Model 150 four Channel Sanborn Recorder. In summary, the signals available were dynamic forcing function, actuator position, actuator position error, actuator force, load inertia position, load inertia acceleration, load friction force, and torque motor current.

Steady-state actuator position versus command input were read from a Model 3500 Electro Instruments Digital Voltmeter and tabulated.

Both static and dynamic gas consumption were read visually on a Daniel sharp edged-orifice flow meter and tabulated. The dynamic flow varied from a maximum in one actuator direction and back to a minimum and then to a maximum again when the actuator moved in the opposite direction and back to a minimum again. The peak readings were read and recorded. The minimums were assumed to correspond to the static reading.

4.3.2.7 Closed Loop Performance

Table 4-4 is a summary of the closed loop performance of actuators S/N-1 and S/N-2 except for snubbing which is tabulated here for convenience.

GAS TEMPERATURE, MEDIUM, AND PRESSURE	ACTUATOR NO.	FIG. NO.	LOAD FRICTION	TRANSIENT RESPONSE (INPUT-0.087" STEP) RISE TIME ①	OVER-SHOOT	SETTLING TIME ②	DYNAMIC RESOLUTION (INPUT-0.0174 IN/SEC. RAMP)	STATIC RESOLUTION (INPUT-0.1 CPS SINE WAVE)	FREQUENCY RESPONSE (INPUT-0.05" SIN ωt) PHASE SHIFT @ 1.0 C.P.S.	MAX. AMPLITUDE RATIO & FREQ.	AMPLITUDE RATIO @ 8 C.P.S.	MAXIMUM VELOCITY ③ (INPUT-SATURATING COMMAND SIGNAL)	SNUBBER ④ DEFLECTION INPUT-MAX.VEL RETRACT	EXTEND	GAS ⑤ CONSUMPTION (INPUT-0.501" SIN 2π(0.3)t)	GAS CONSUMPTION (INPUT-STEADY-STATE POSITION)
—	—	—	POUNDS	SEC.	%	SEC	± IN.	± IN.	DEG.	IN./IN. @ C.P.S.	DB	IN/SEC	IN.	IN.	LB./MIN.	LB./MIN.
-270°F HYDROGEN @ 800 PSIG	S/N-2	4-67, 68, 69	SEE FIGURES	0.038	8.7	0.45	± 0.007	± [illegible]	-27	0.66 @ 1	-3.85	1.59	FIGURE 4-91		NO DATA	NO DATA
		4-70, 71, 72	0	0.06	17.7	0.26	± 0.007	± 0.011	-27	1.34 @ 4	-1.2	2.63	0.0385	0.0514	—	—
R.T. HYDROGEN @ 800 PSIG	S/N-2	4-73, 74, 75	SEE FIGURES	0.04	8.7	0.25	± 0.0025	± 0.01	-15	0.72 @ 5	-9.6	3.1	—	—	1.52	0.71
		4-76, 77, 78	0	0.045	32.8	0.39	± 0.007	± 0.005	-16.6	1.66 @ 6	+3.3	3.94	—	—	—	—
R.T. NITROGEN @ 800 PSIG	S/N-2	4-79, 80, 81	SEE FIGURES	0.04	[illegible]	0.20	± 0.004	± 0.013	-32.4	1.49 @ 10	+2.8	1.59	—	—	7.98	2.65
		4-82, 83, 84	0	0.06	10.0	0.40	± 0.007	± 0.01	-25	1.18 @ 6	+0.34	2.0	—	—	—	—
R.T. HYDROGEN @ 800 PSIG	S/N-1	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		4-85, 86, 87	0	0.04	24	0.65	± 0.007	± 0.011	-26	1.81 @ 4	-3.6	NO DATA	—	—	NO DATA	NO DATA
-250°F HYDROGEN @ 800 PSIG	—	—	7,300	0.0419	≦30	0.35	—	—	≦20	1.5	-3 DB RELATIVE TO THE LOW FREQUENCY AMPLITUDE RATIO	—	≦0.1	≦0.1	1.5 LB/MIN. MAX.	0.5 LB/MIN. MAX.

NOTES:
① TIME TO TRAVEL FROM 10 TO 90% OF FINAL DISPLACEMENT.
② TIME TO SETTLE TO WITHIN 2 TO 5% OF FINAL POSITION.
③ ACTUATOR DOES NOT HAVE A CONSTANT FORCE LOAD.
④ SNUBBING IS AN OPEN LOOP TEST— TORQUE MOTOR IS SATURATED.
⑤ RESULTS ARE CALCULATED - BASED ON ASSUMED SINE WAVE VARIATION, STEADY STATE FLOW, AND VISUAL PEAK READINGS OF (H_2 +Λ 1.39, S.S. 0.71, -Λ 1.14 LB/MIN, N_2 +Λ 8.71, S.S. 2.65, -Λ 8.05 LB/MIN.)

Table 4-4. Summary Of Actuators S/N-1 And S/N-2 Closed Loop Performance Test Results.

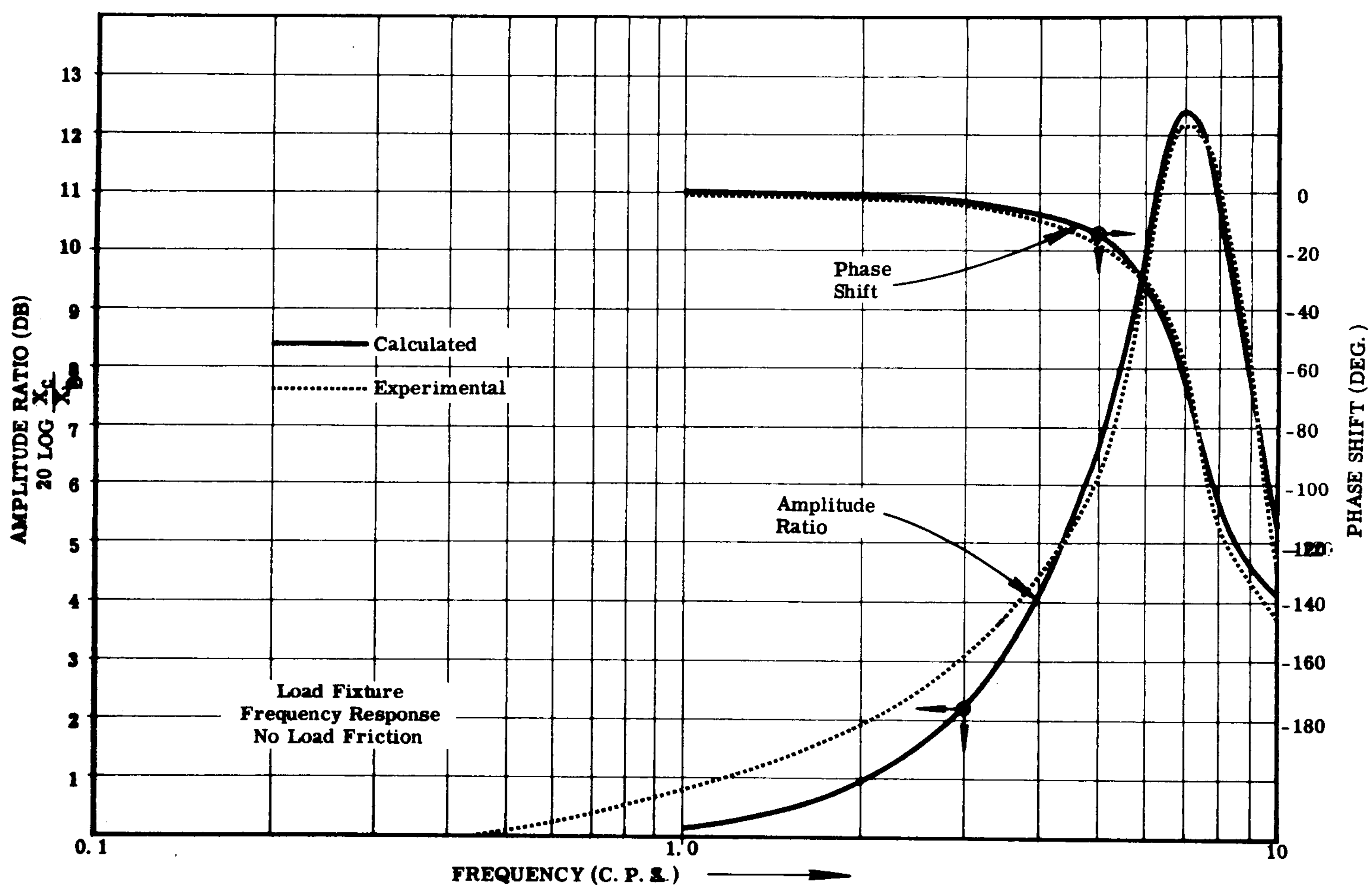


Figure 4-56. Load Fixture Frequency Response (No Load Friction).

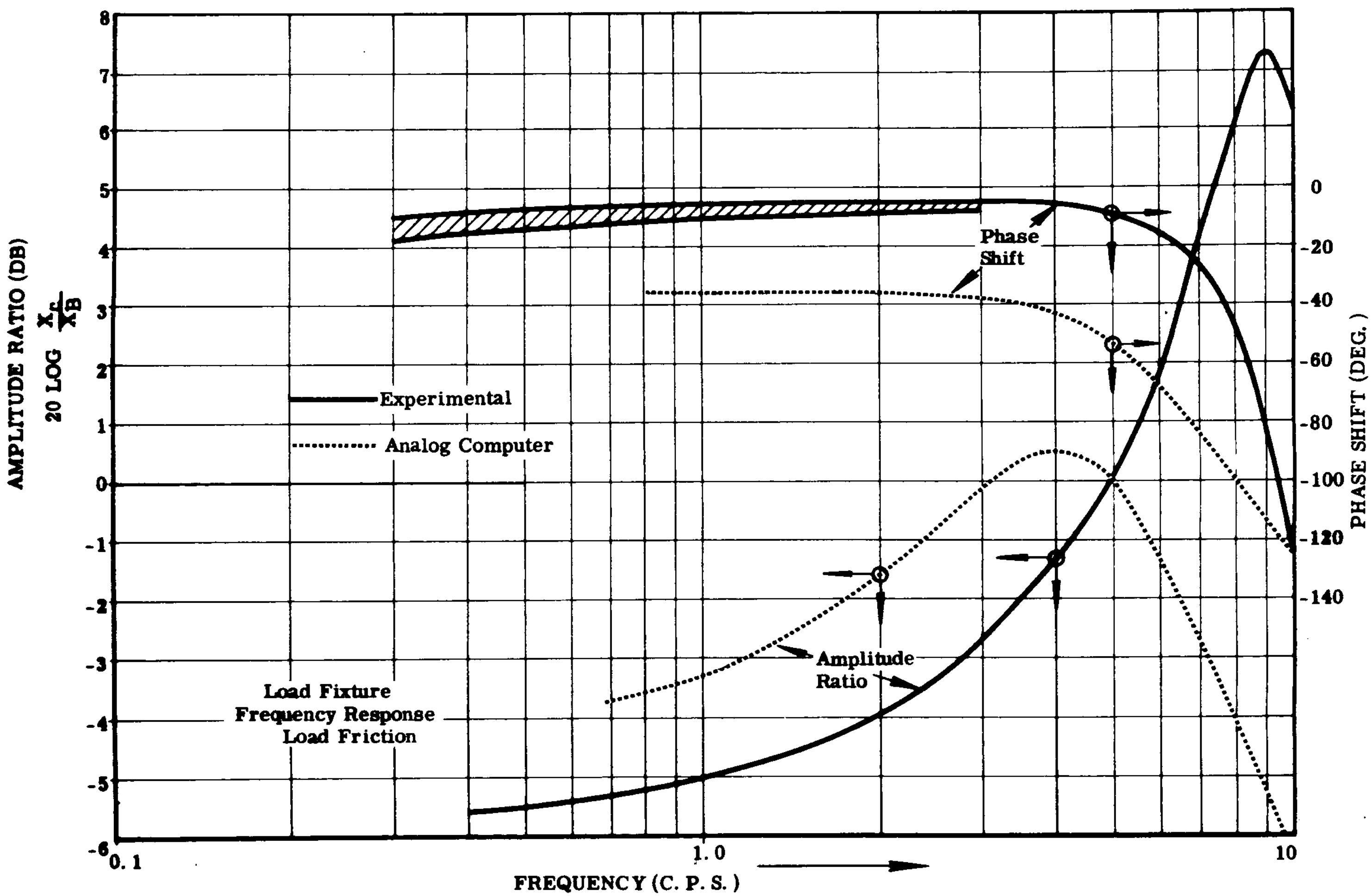


Figure 4-57. Load Fixture Frequency Response (With Load Friction).

Figures 4-58 through 4-61 are the Bode plots of the closed loop frequency responses of actuators S/N-1 and S/N-2.

Figure 4-62 is a no-load closed position calibration curve of actuator S/N-2.

All performance data figures are for actuator S/N-2 unless otherwise noted.

Figures 4-67 through 4-87 are actuators S/N-1 and S/N-2 closed loop performance data from which the values for Table 4-4 and Figures 4-58 through 4-61 were obtained.

4.3.2.8 Open Loop Performance

Figures 4-63 and 4-64 are the acceleration characteristics of actuator S/N-2 with an inertia load and room temperature hydrogen and nitrogen gas respectively.

Figures 4-65 and 4-66 are the constant load velocity characteristics of actuator S/N-2 with room temperature hydrogen and nitrogen gas respectively.

Figures 4-88 through 4-90 are the open loop performance data of actuator S/N-2 from which the values for Figures 4-63 through 4-66 were obtained.

Figure 4-91 is the open loop snubbing data from which the values under snubbing of Table 4-4 were obtained.

4.3.2.9 Discussion Of Results

The control system was evaluated for two load conditions: spring rate, inertia, and load friction; and spring rate, inertia, and no-load friction. Since the performance of the control system is dependent on both the control and the load dynamics and, since the load fixture did not conform exactly to the specifications, an analytical and an experimental definition taken simultaneously with the performance data was deemed necessary.

The experimental load fixture data for no-load friction was taken from the frequency response data and plotted as shown in Figure 4-56. An analytical description was evolved and also plotted in the same figure for comparison. The experimental load fixture data with load friction condition was taken from the frequency response data and plotted as shown in Figure 4-57. Because the load fixture friction brake force did not remain constant, and the brake developed deadband during testing, an analytical description was not attempted. Instead an analog computer description obtained from a preliminary control system analysis with the specified load was plotted in the same figure for comparison.

(Text continued on page 4-116)

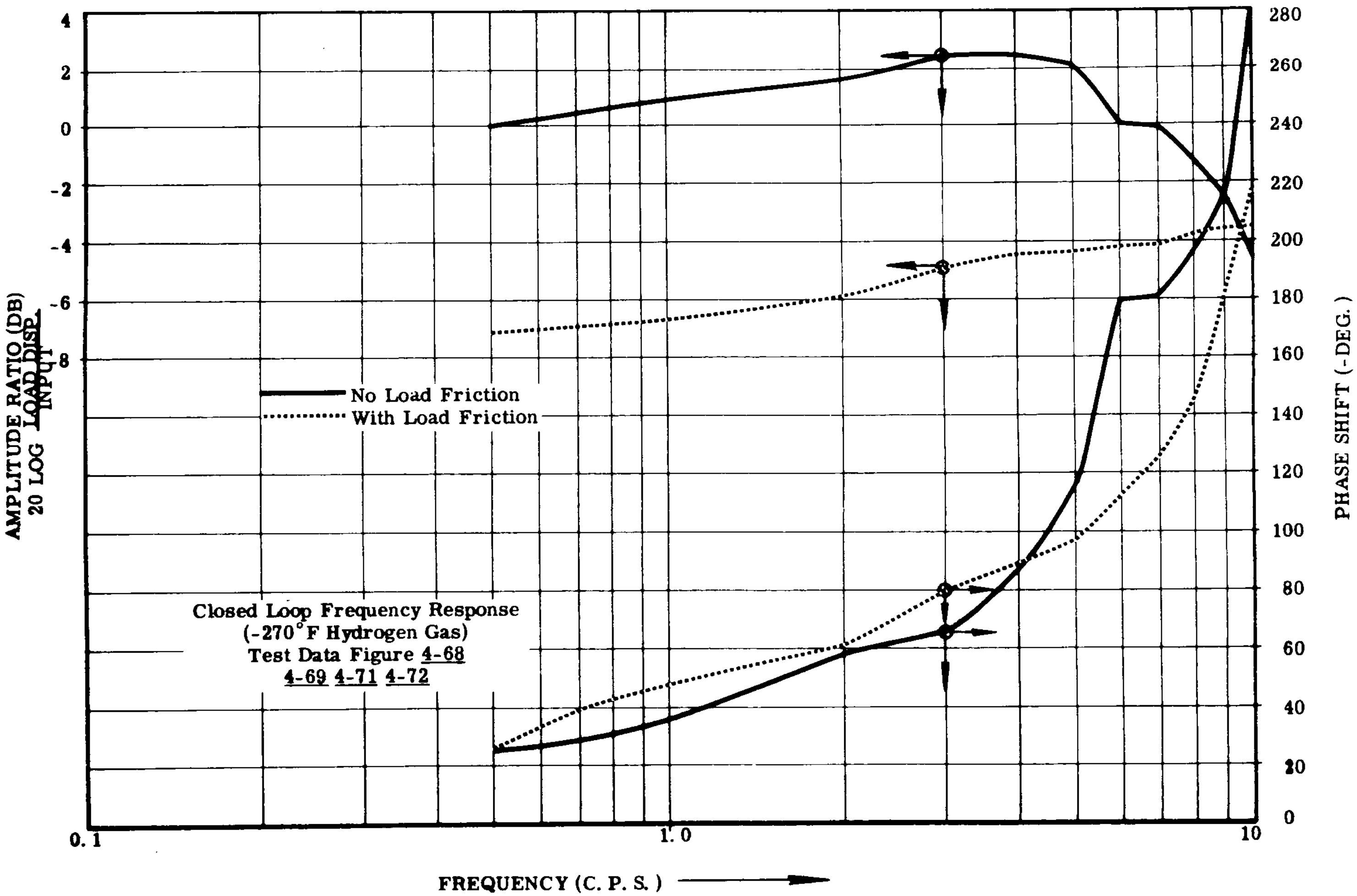


Figure 4-58. Actuator S/N 2 Closed Loop Frequency Response (-270° H_2).

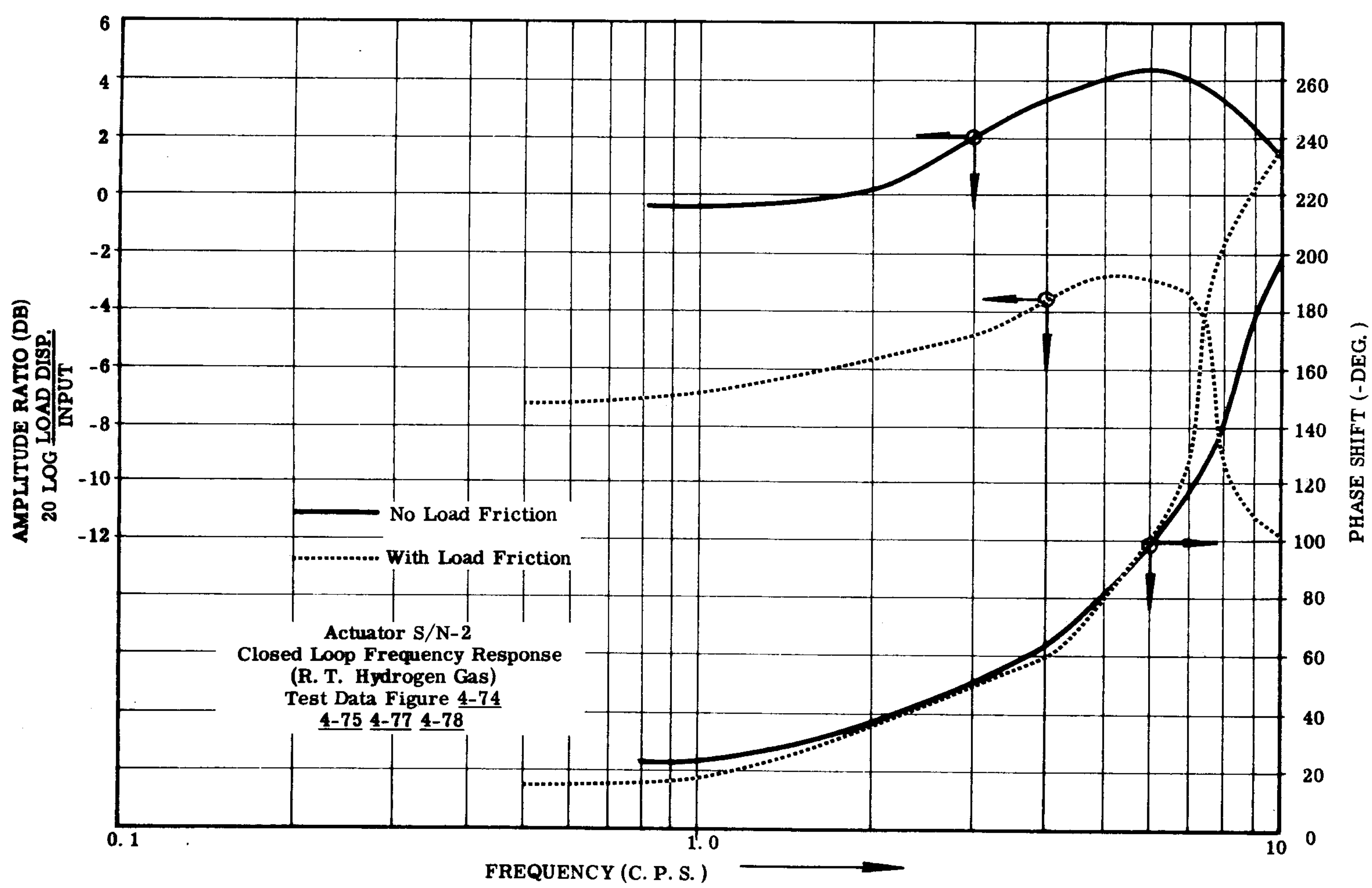


Figure 4-59. Actuator S/N 2 Closed Loop Frequency Response (Room Temperature H_2).

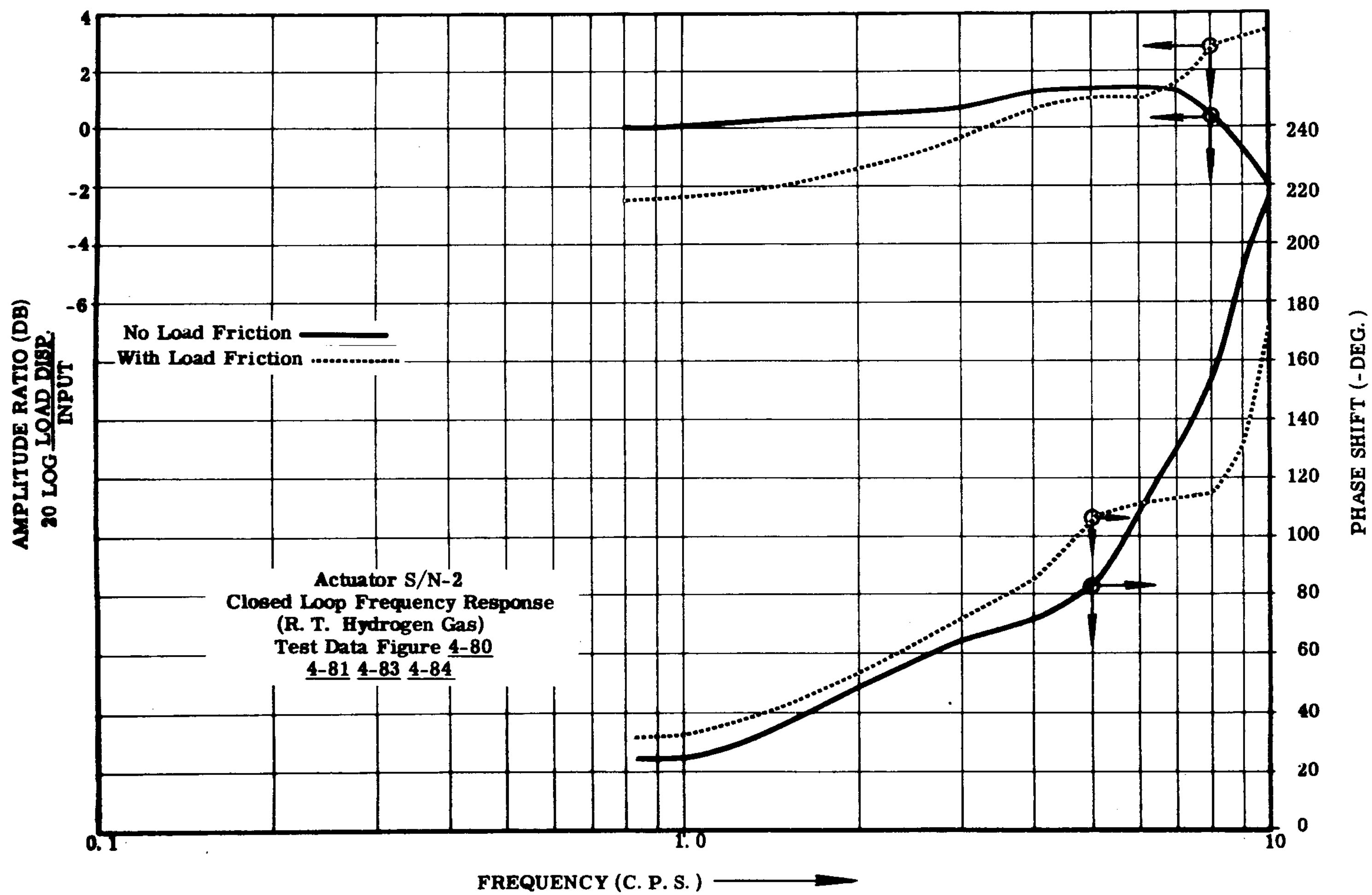


Figure 4-60. Actuator S/N 2 Closed Loop Frequency Response (Room Temperature N_2).

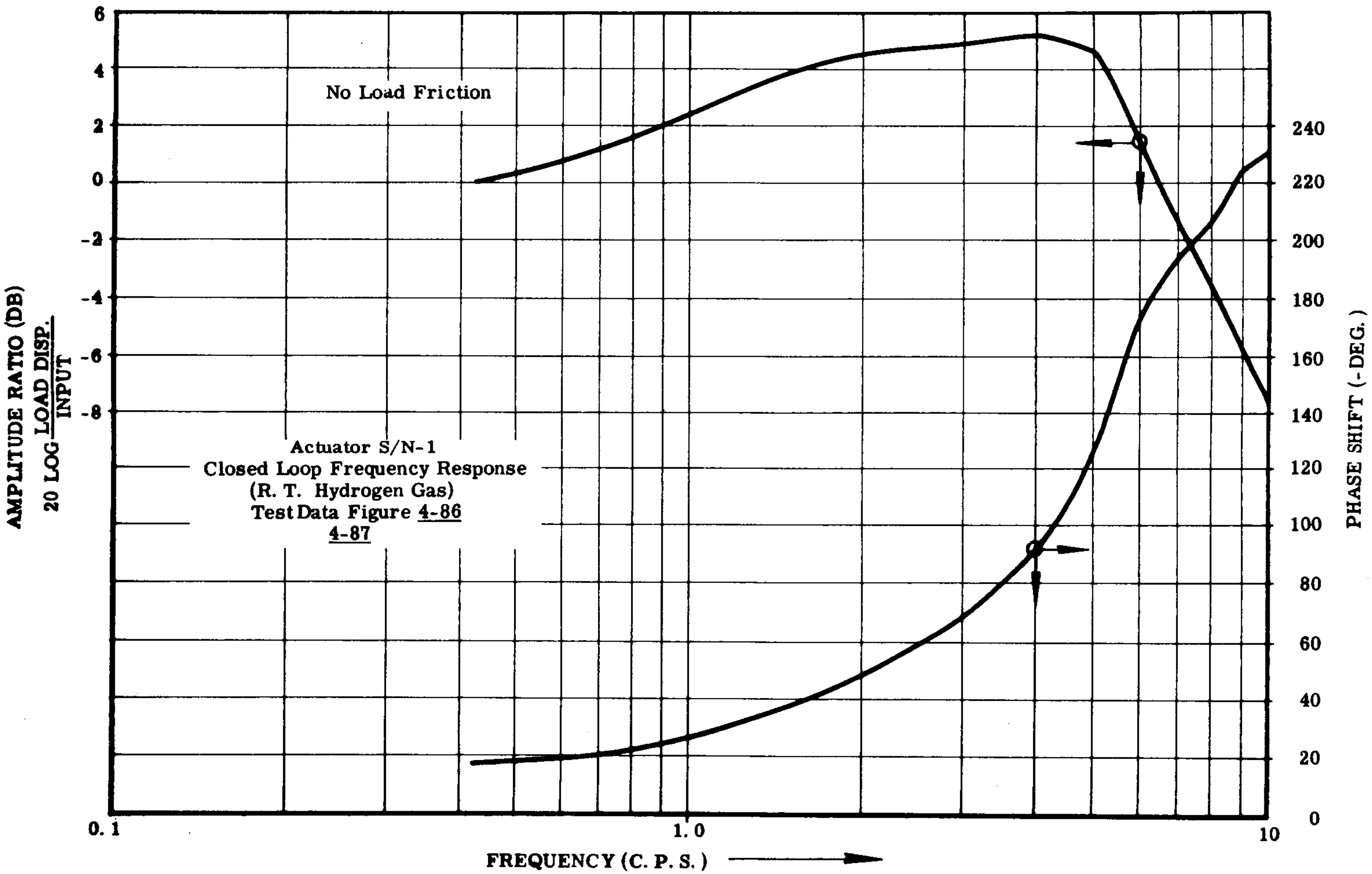


Figure 4-61. Actuator S/N 1 Closed Loop Frequency Response (Room Temperature N_2).

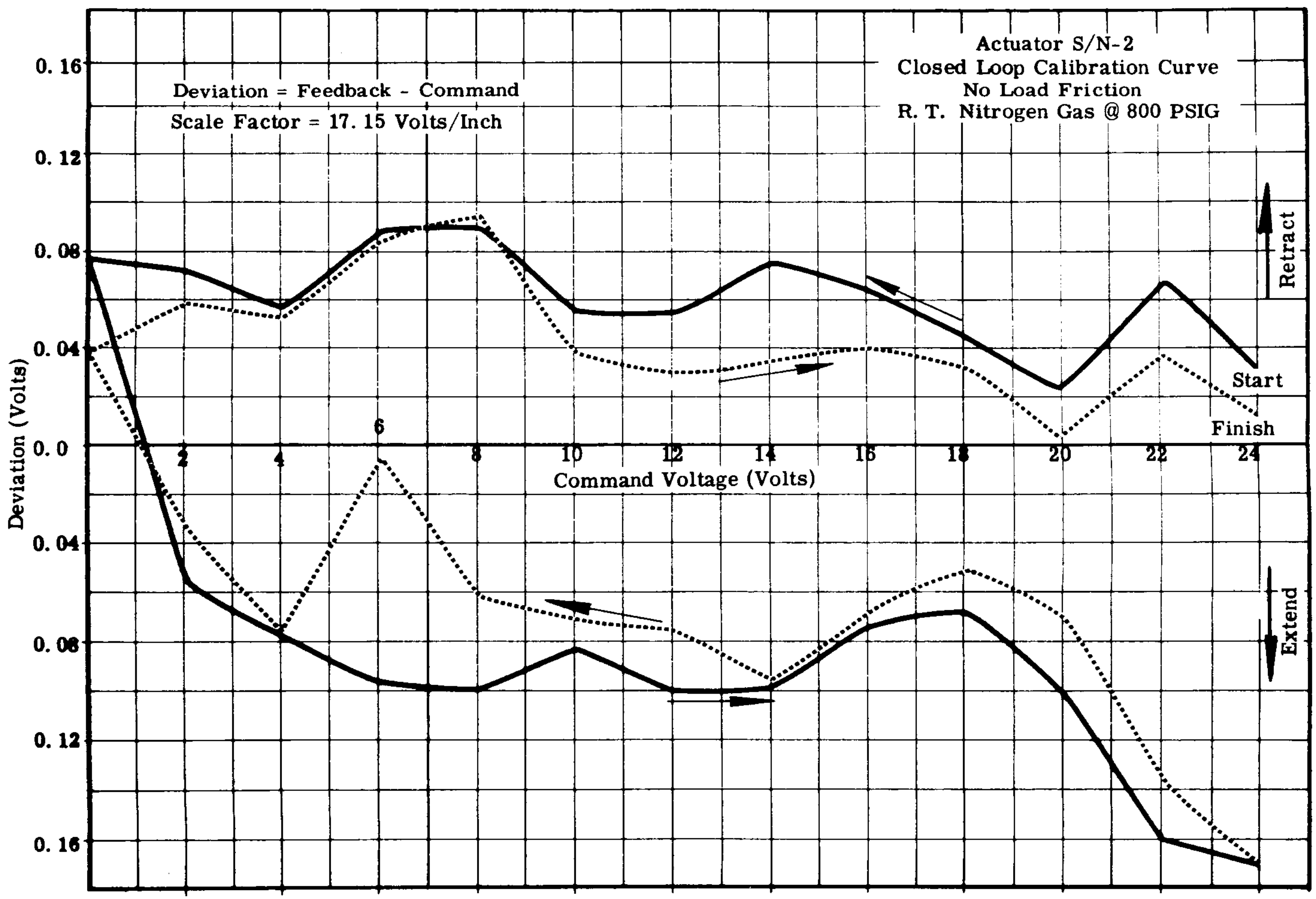


Figure 4-62. Actuator S/N 2 Closed Loop Calibration Curve.

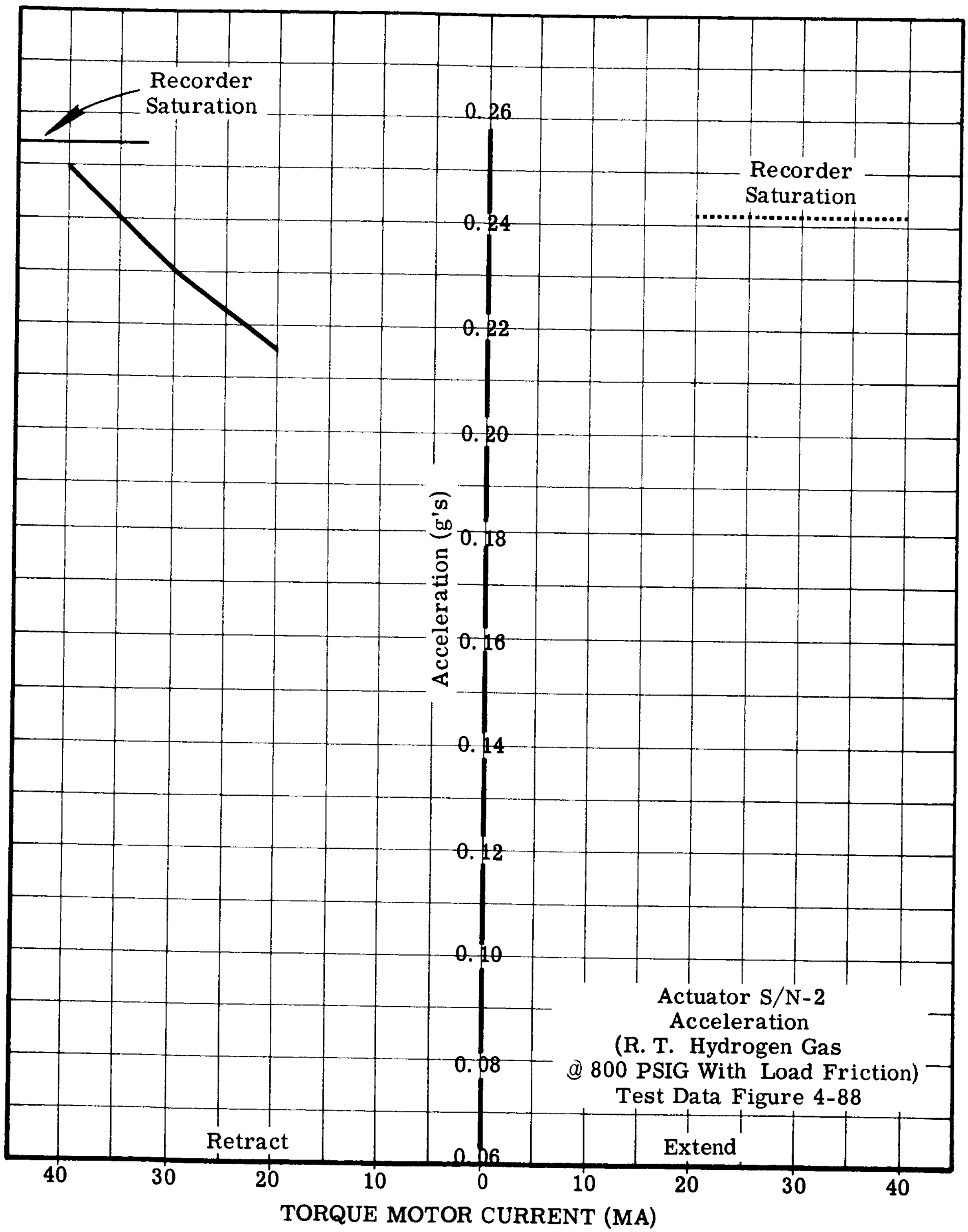


Figure 4-63. Actuator S/N 2 Acceleration (Using H_2).

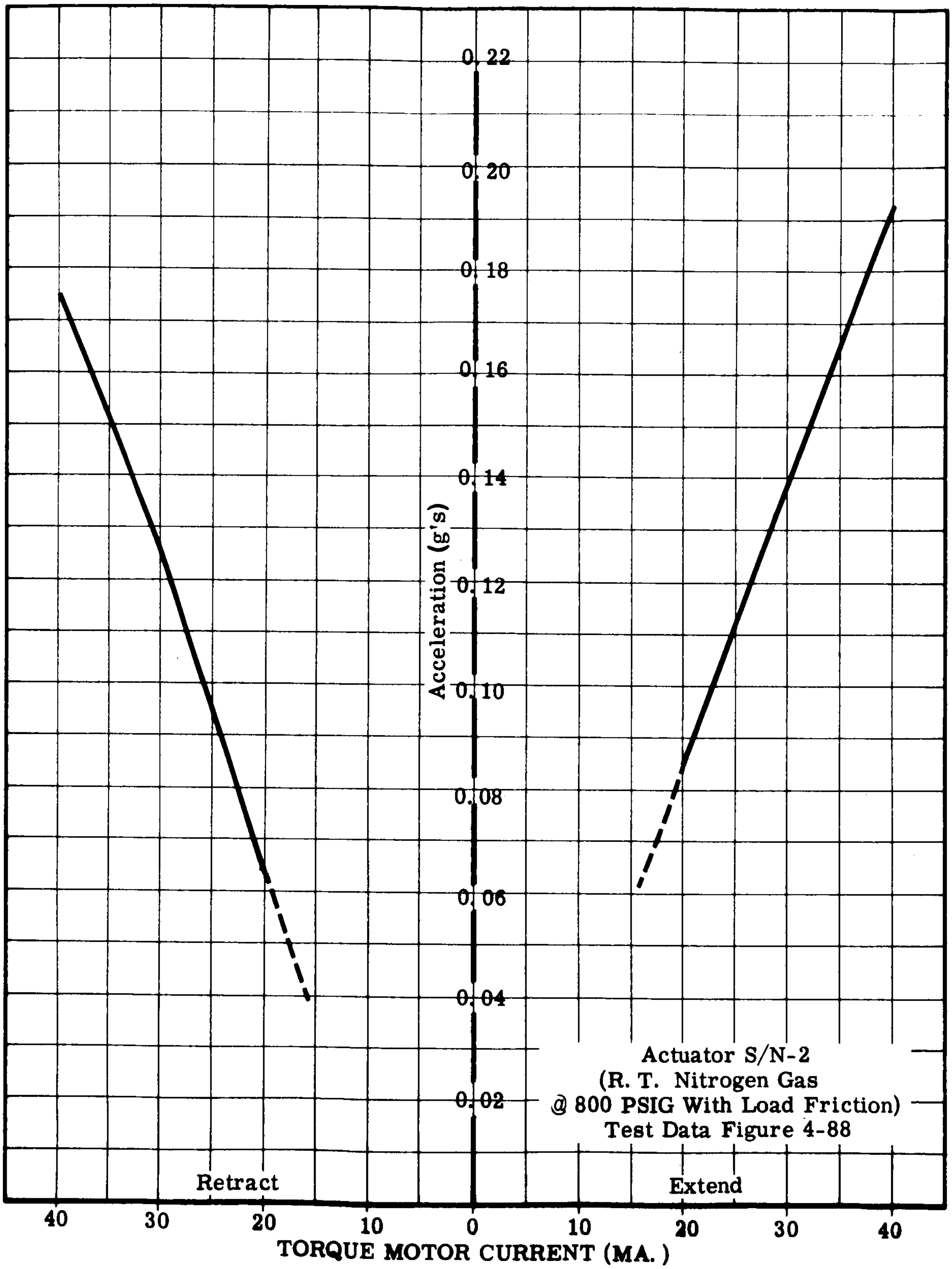


Figure 4-64. Actuator S/N 2 Acceleration (Using N_2).

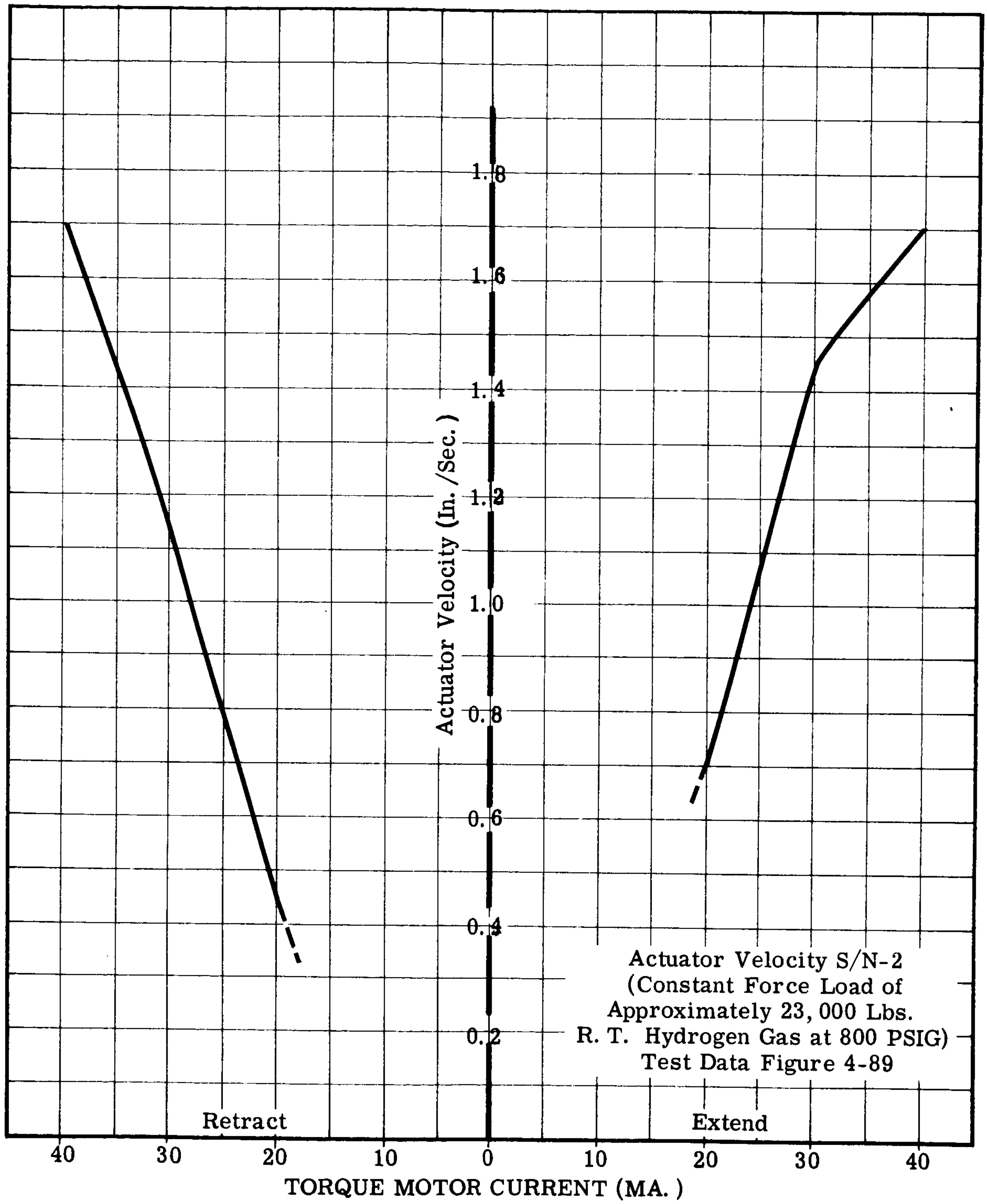


Figure 4-65. Actuator S/N 2 Velocity (Using H_2).

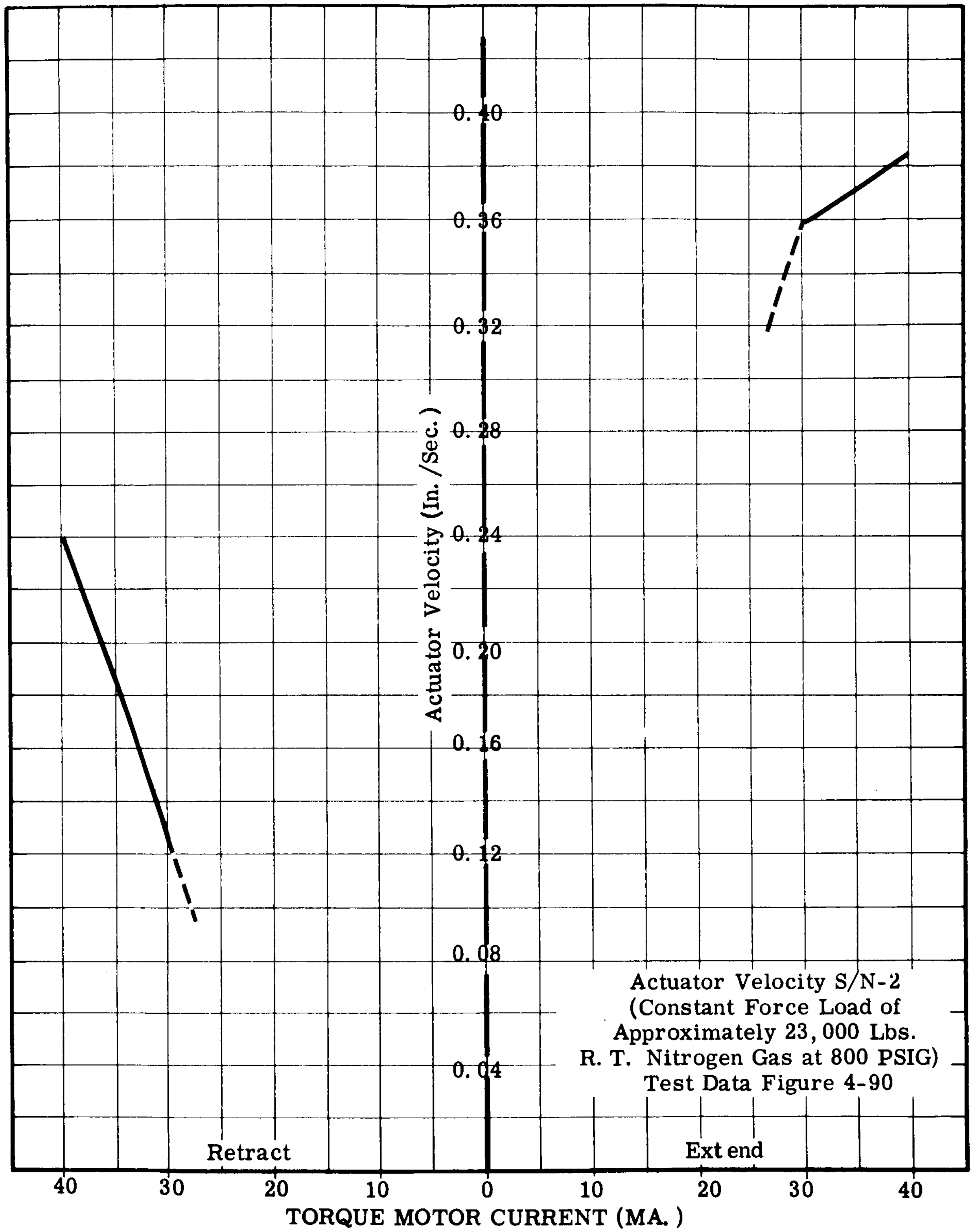


Figure 4-66. Actuator S/N 2 Velocity (Using N_2).

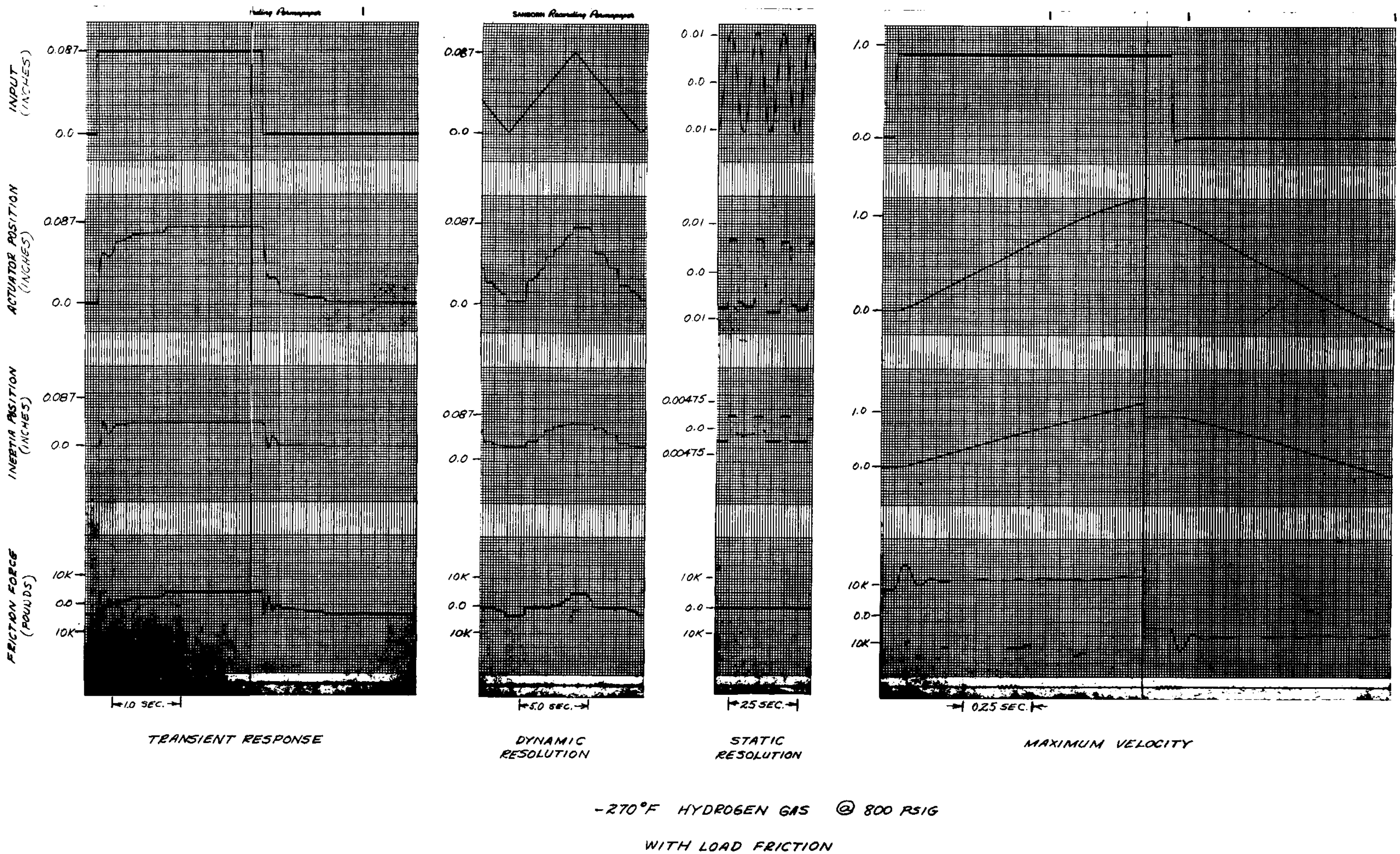


Figure 4-67. Actuator Closed Loop Performance.

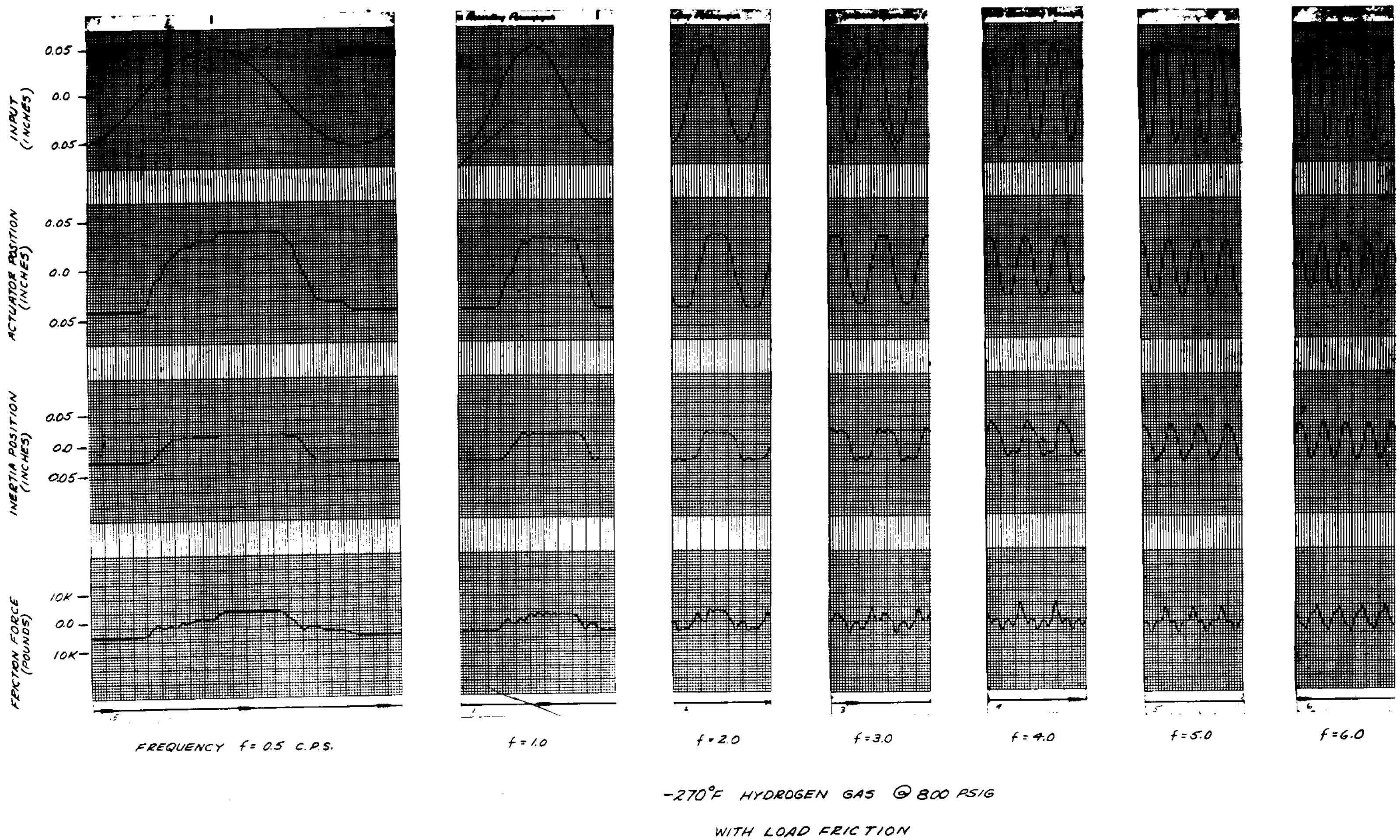


Figure 4-68. Actuator Closed Loop Frequency Response.

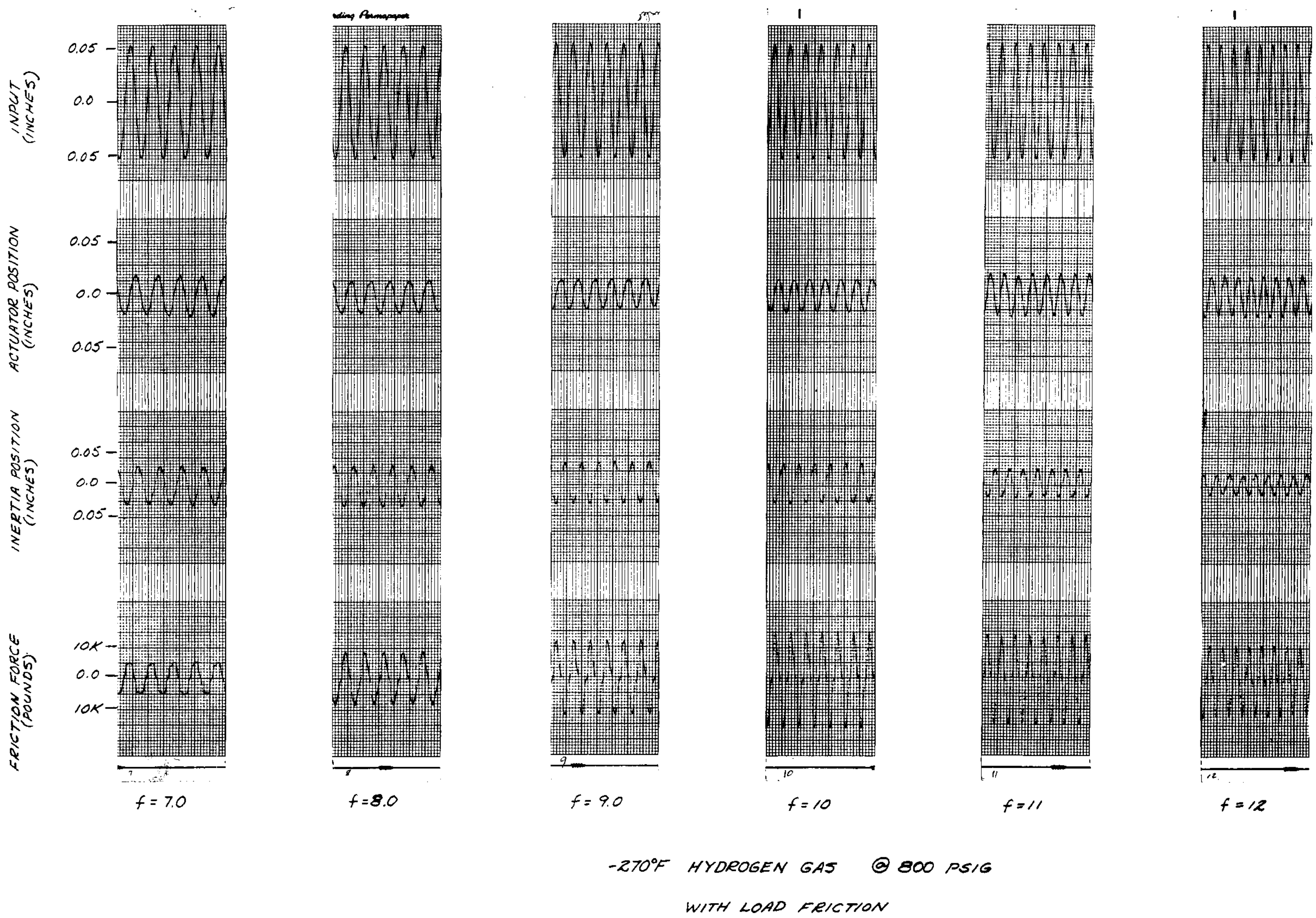


Figure 4-69. Actuator Closed Loop Frequency Response.

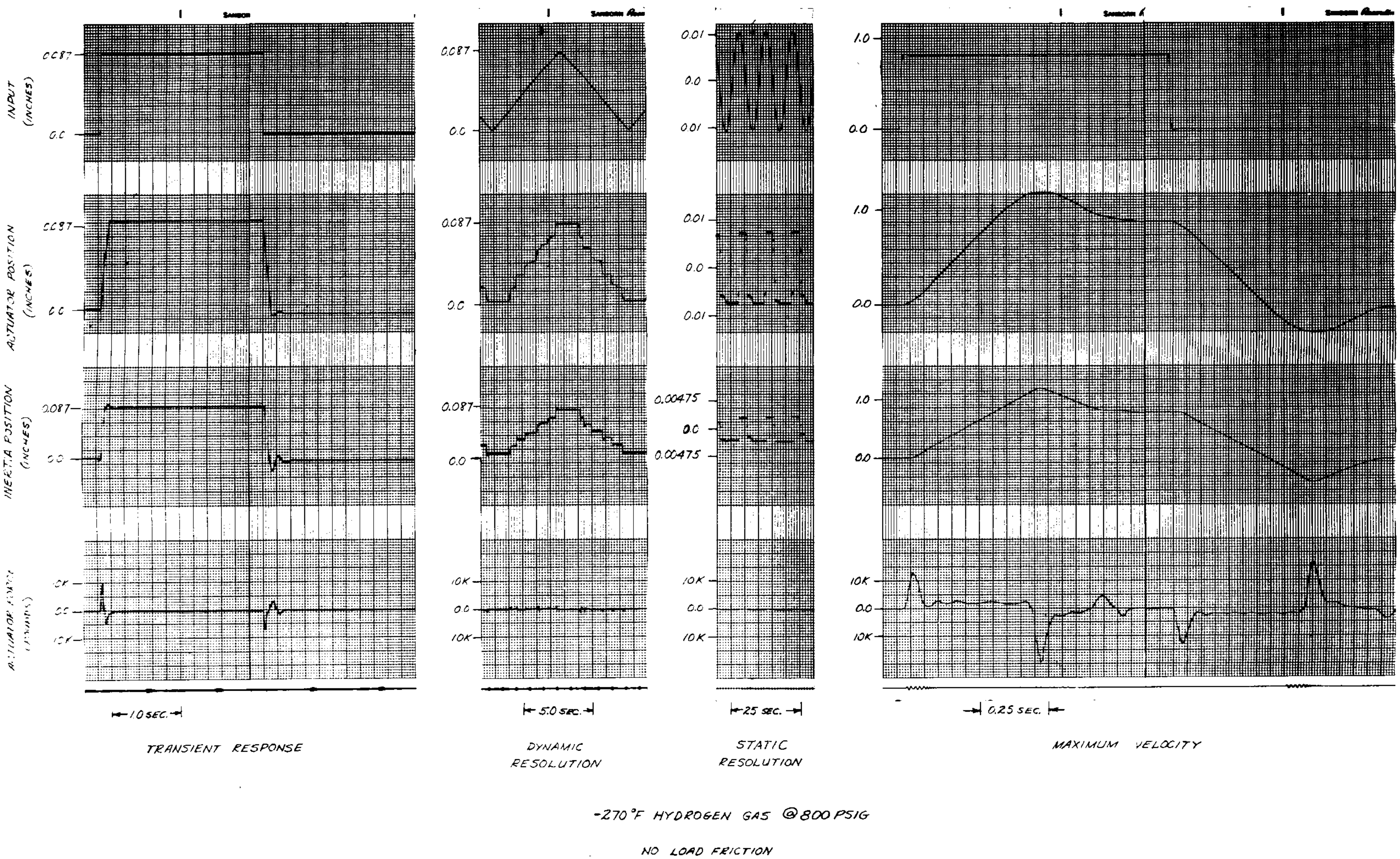


Figure 4-70. Actuator Closed Loop Performance.

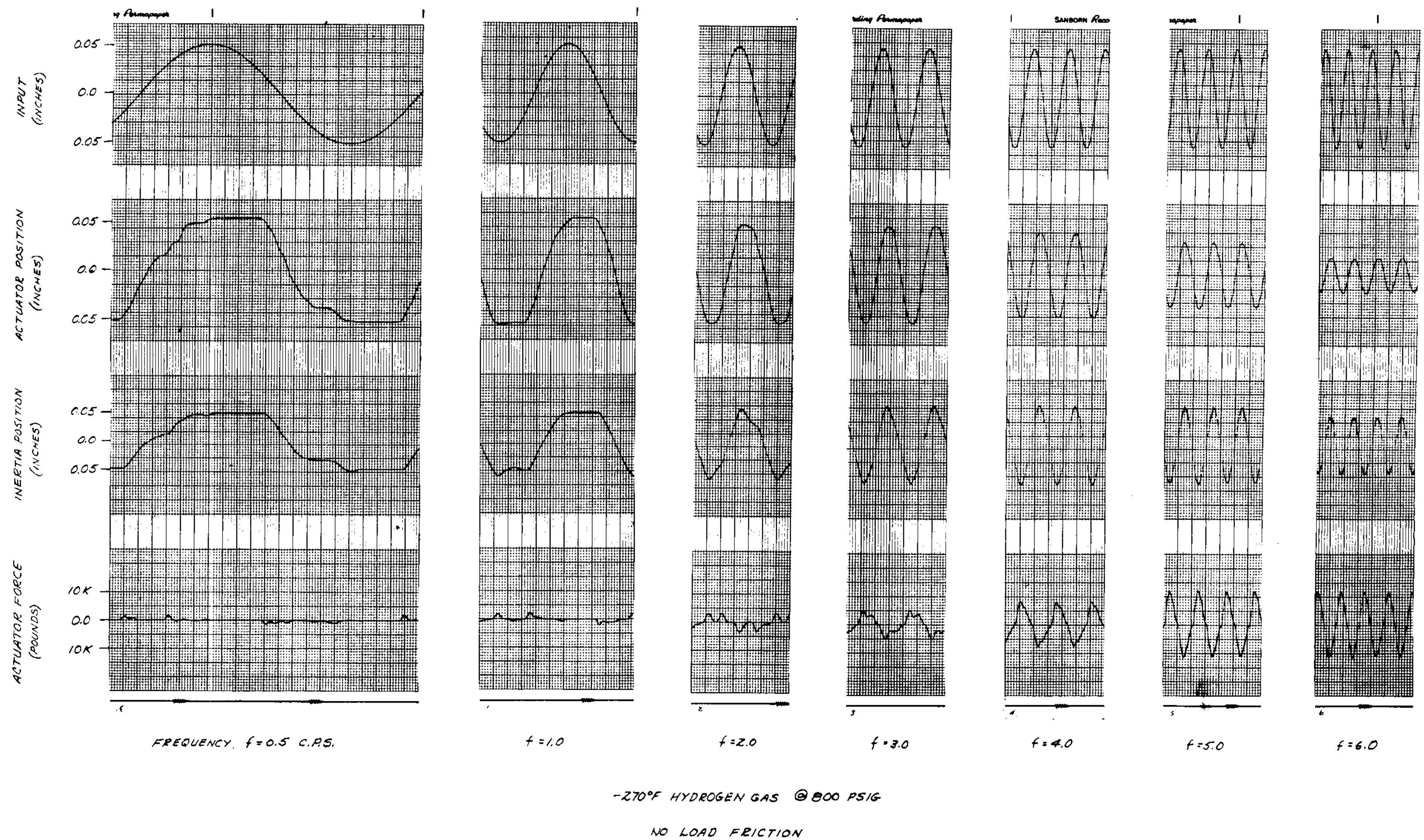


Figure 4-71. Actuator Closed Loop Frequency Response.

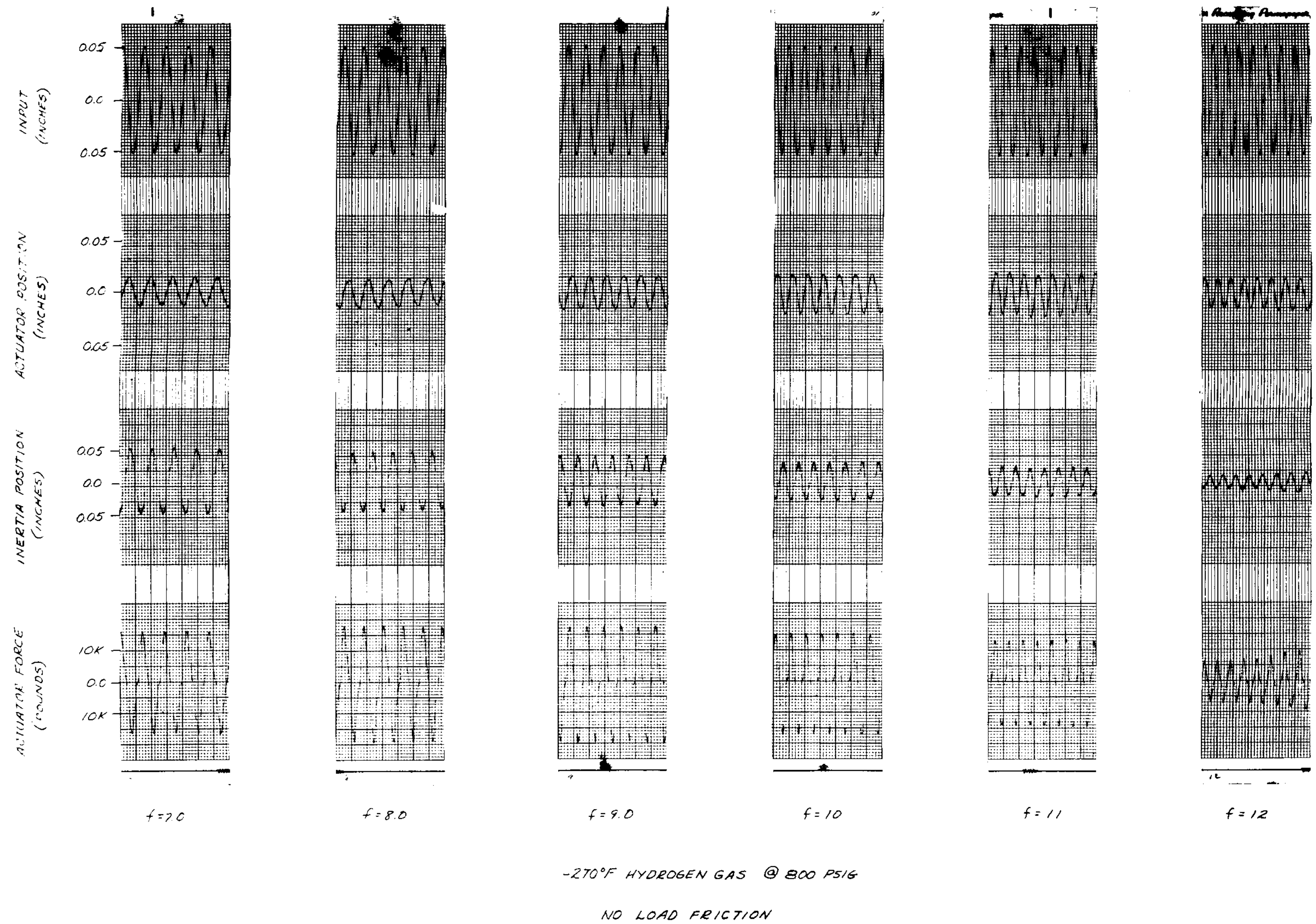


Figure 4-72. Actuator Closed Loop Frequency Response.

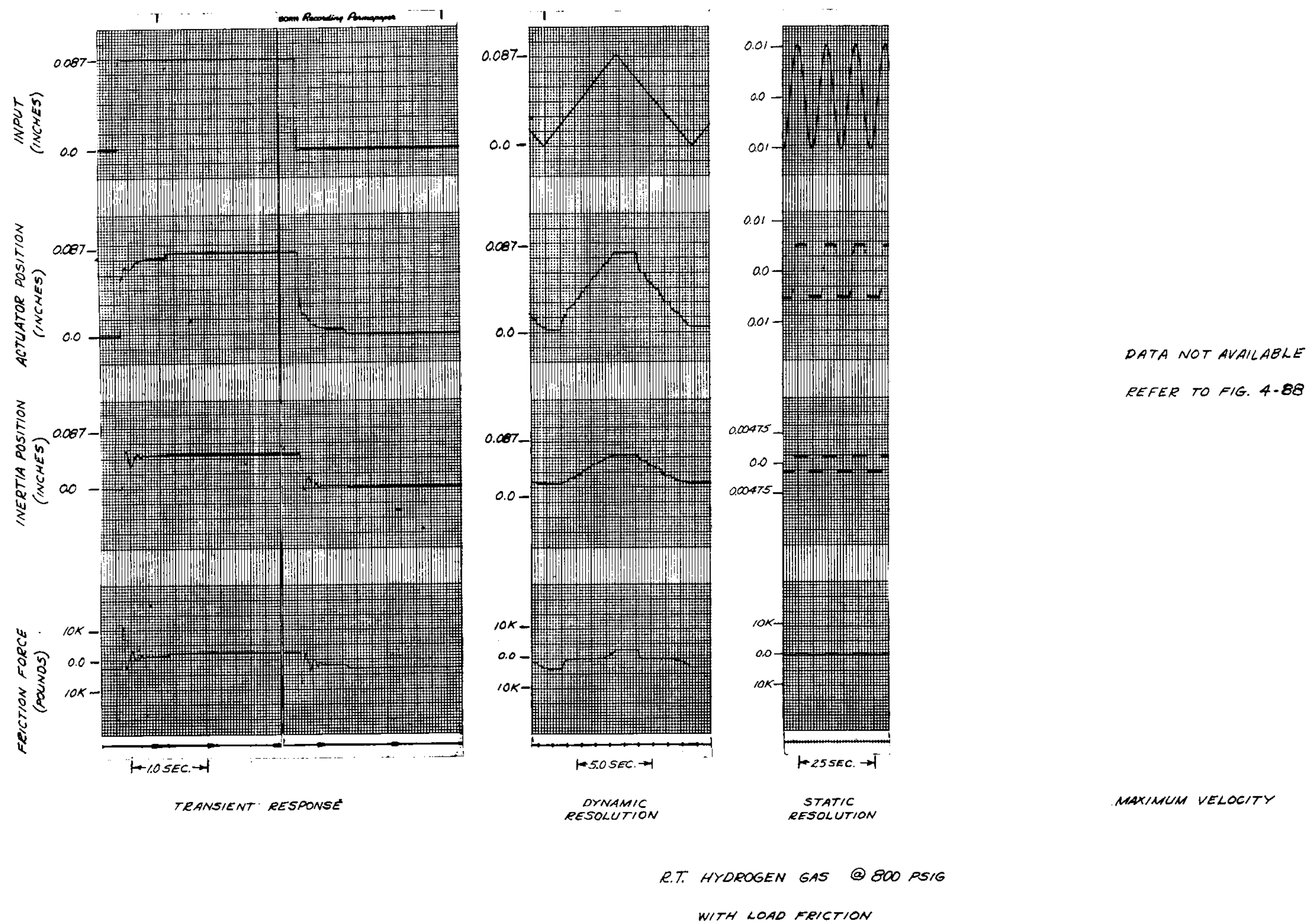


Figure 4-73. Actuator Closed Loop Performance.

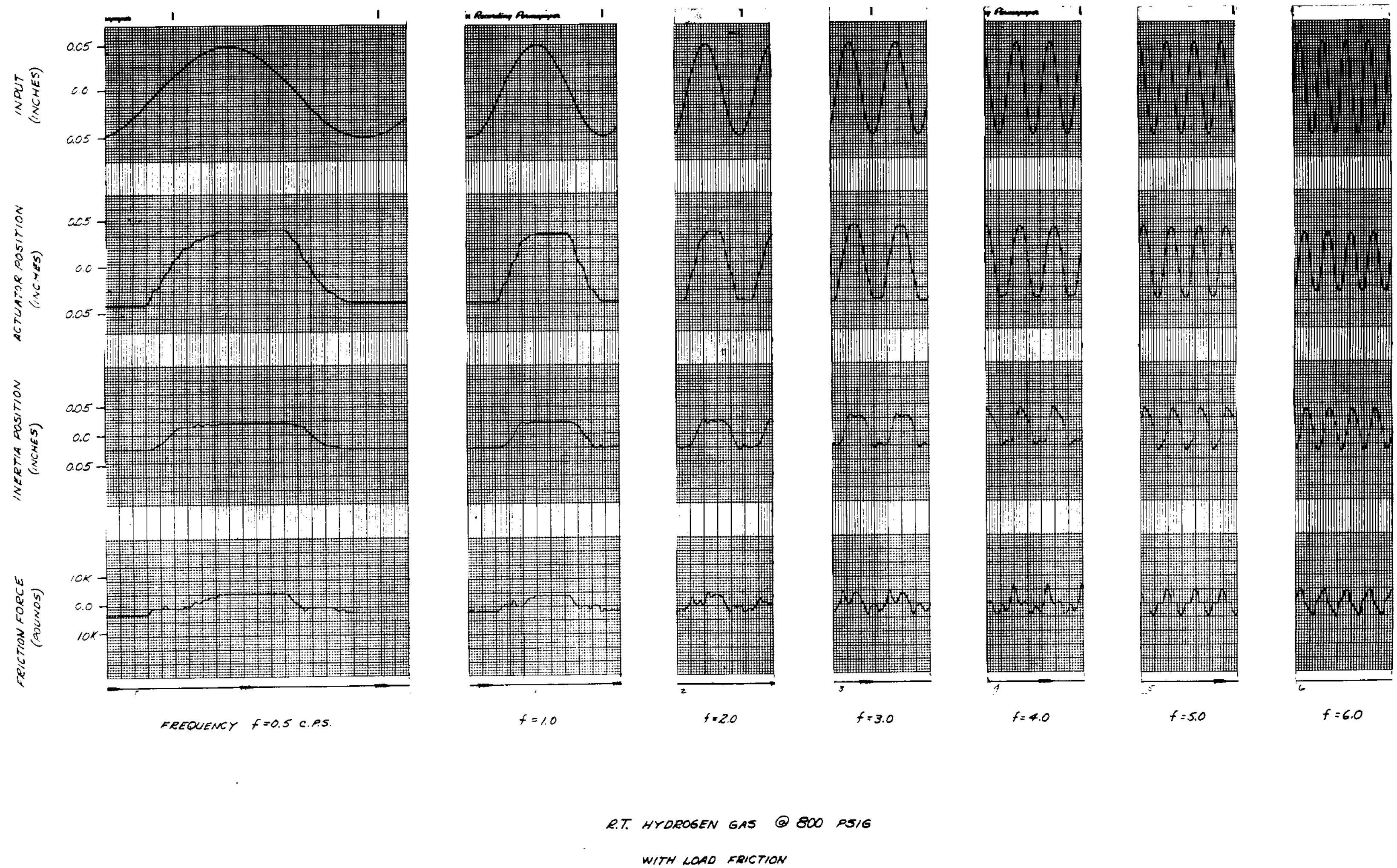


Figure 4-74. Actuator Closed Loop Frequency Response.

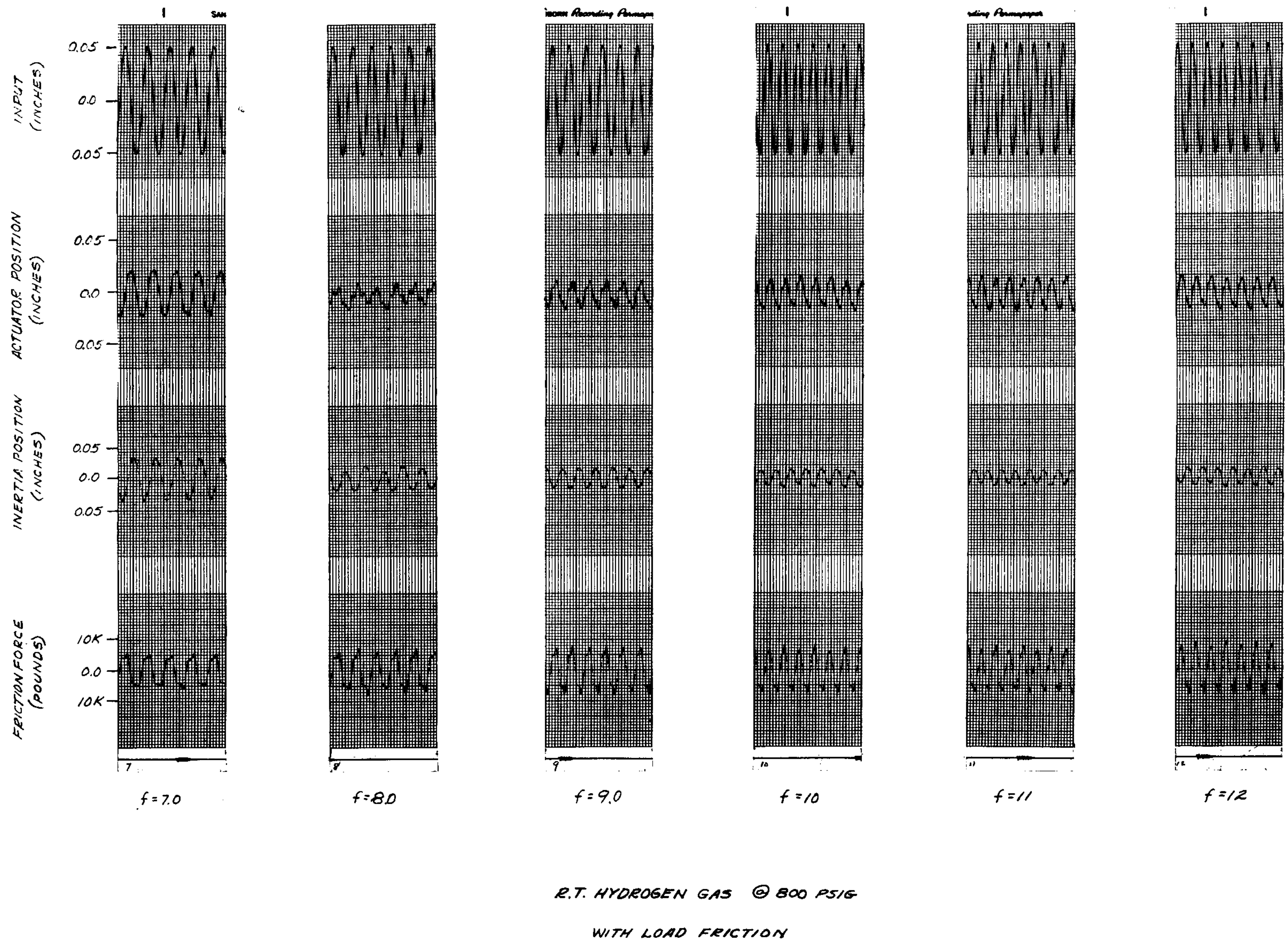


Figure 4-75. Actuator Closed Loop Frequency Response.

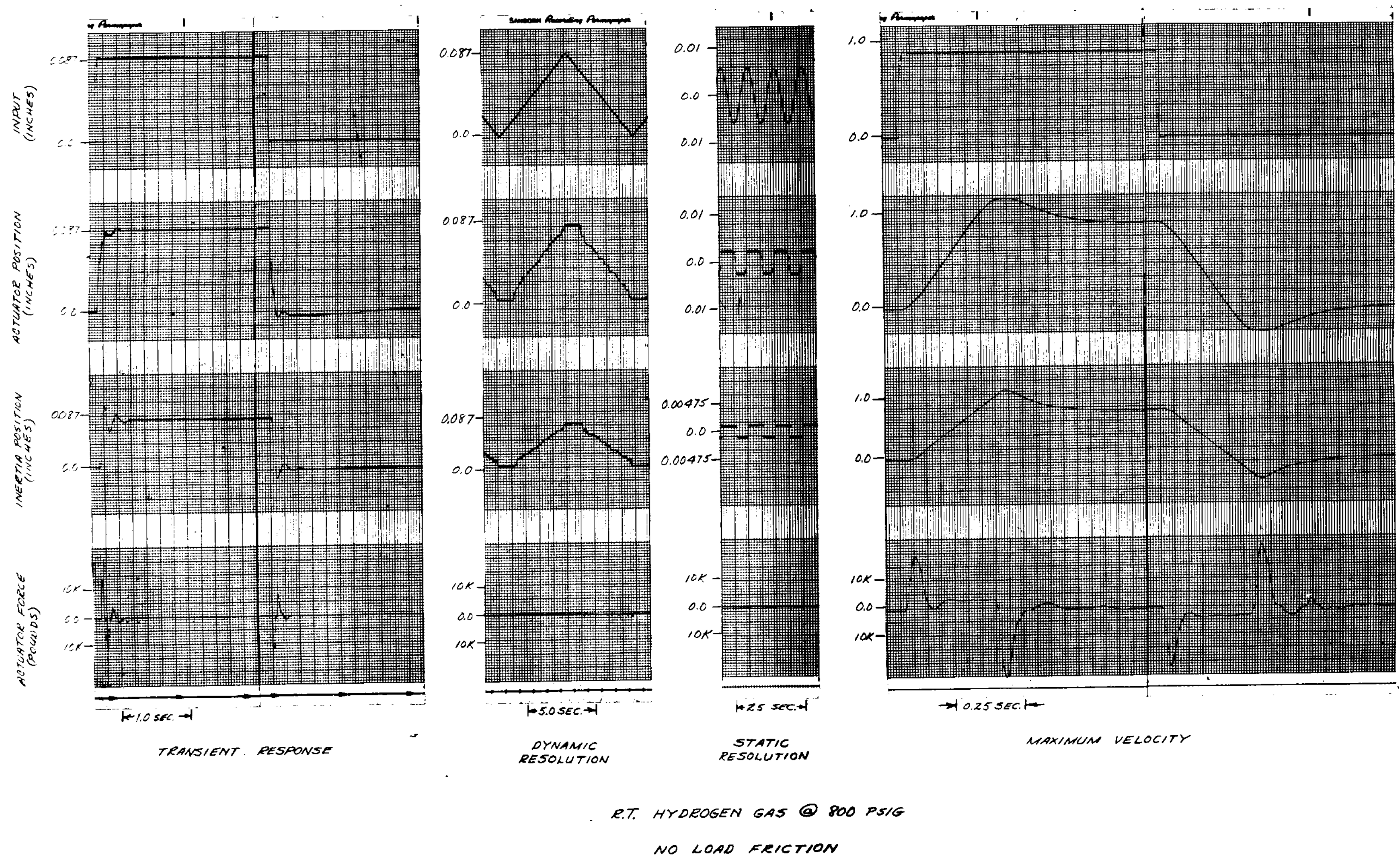
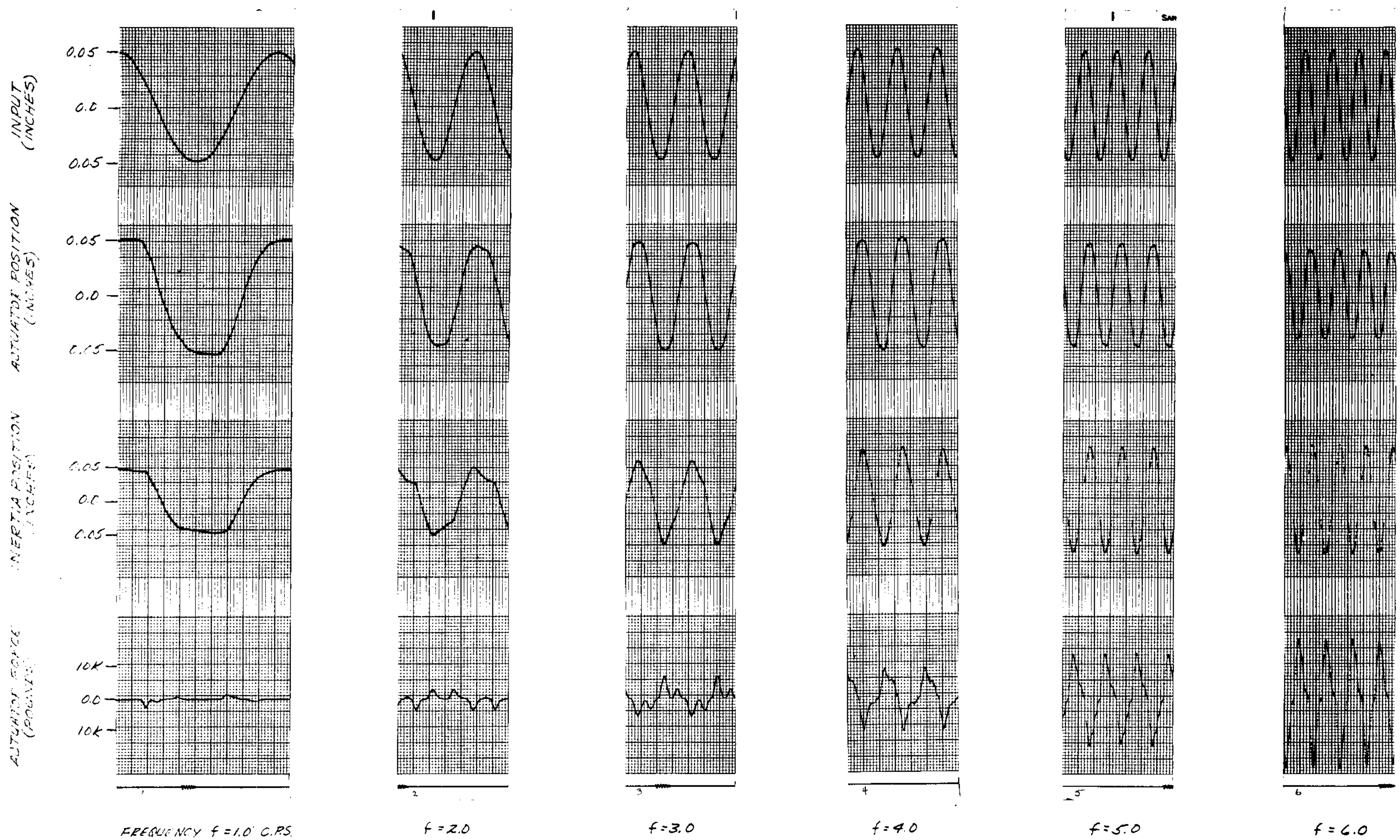


Figure 4-76. Actuator Closed Loop Performance.



R.T. HYDROGEN GAS @ 800 PSIG

NO LOAD FRICTION

Figure 4-77. Actuator Closed Loop Frequency Response.

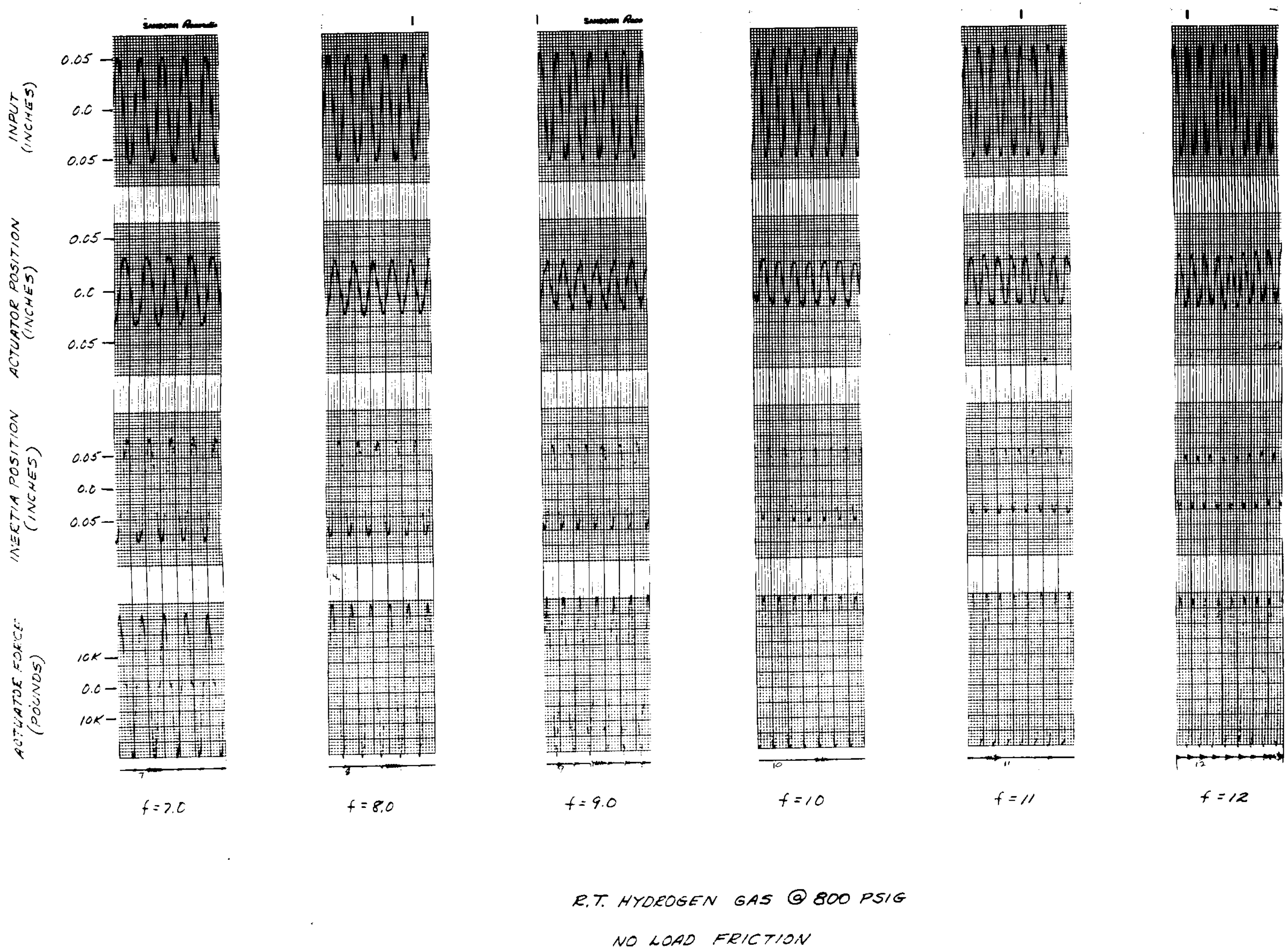


Figure 4-78. Actuator Closed Loop Frequency Response.

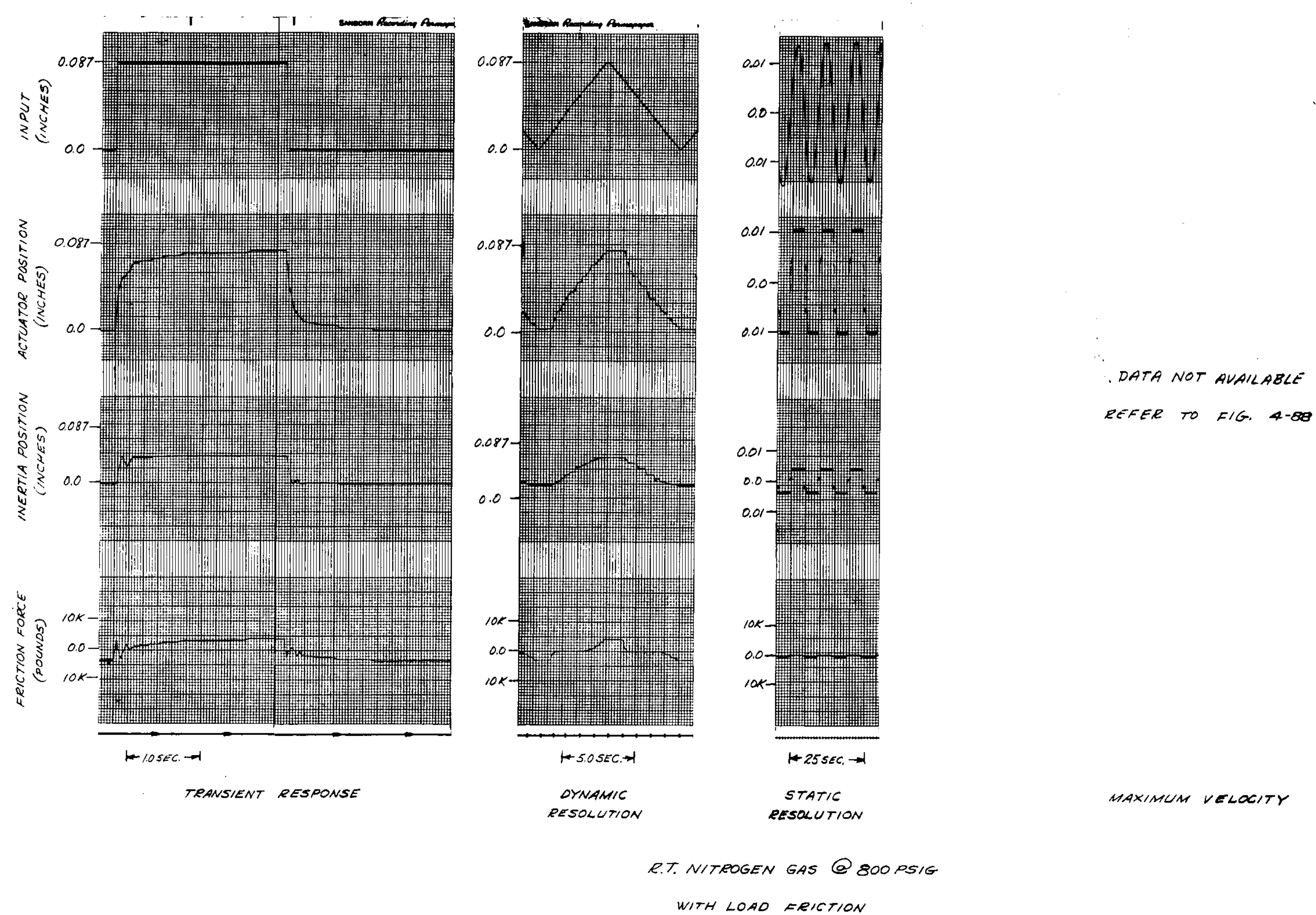


Figure 4-79. Actuator Closed Loop Performance.

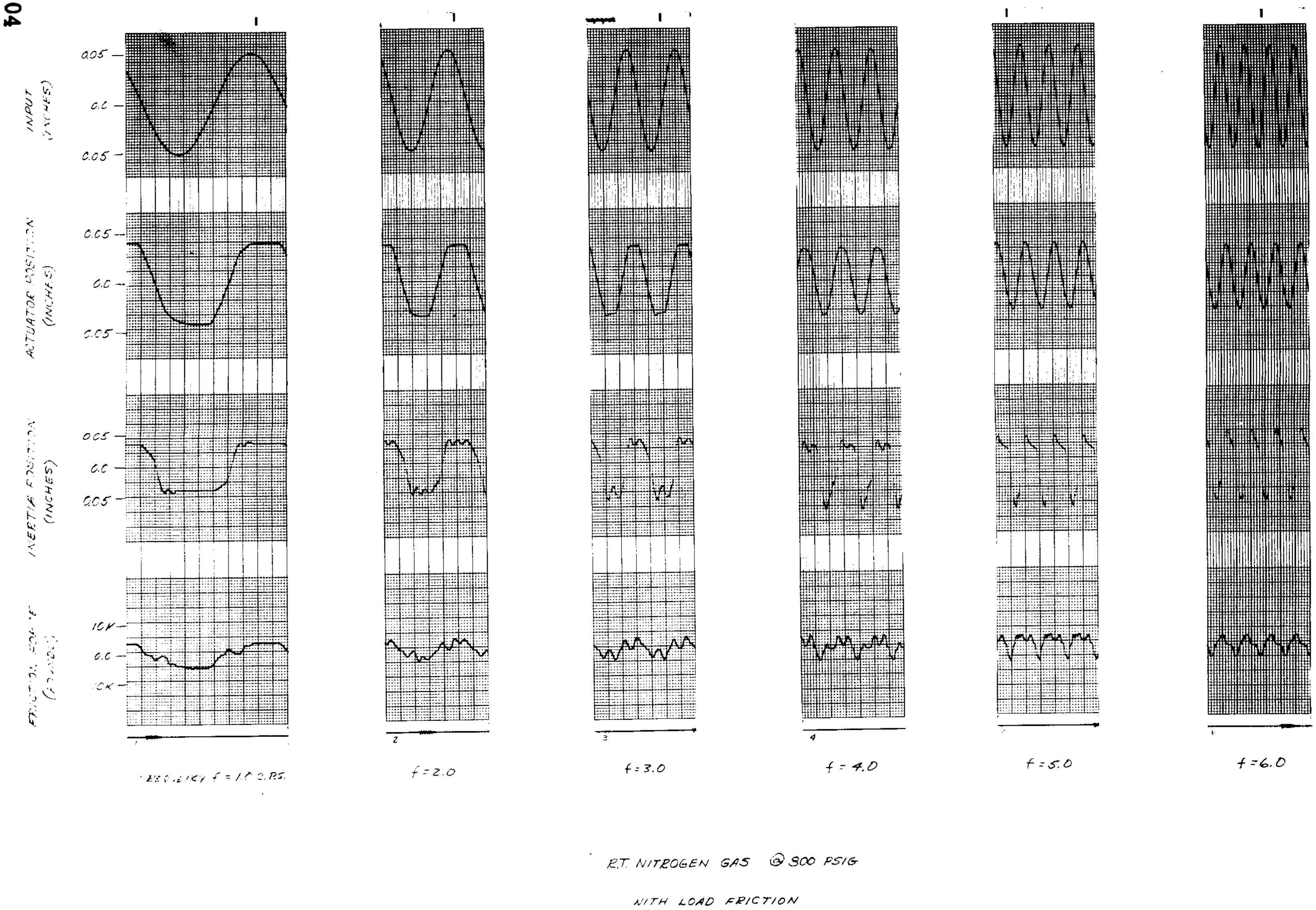


Figure 4-80. Actuator Closed Loop Frequency Response.

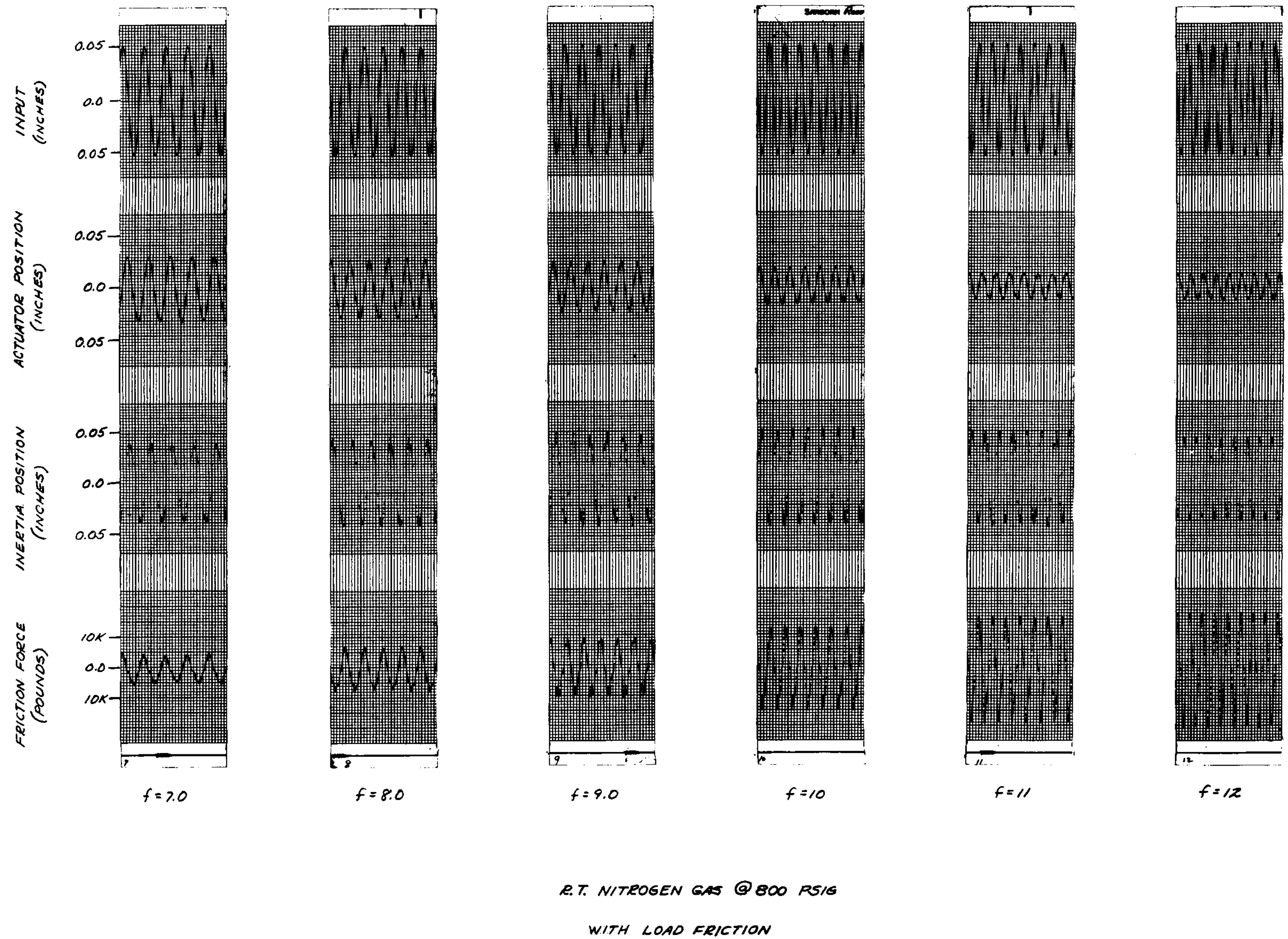


Figure 4-81. Actuator Closed Loop Frequency Response.

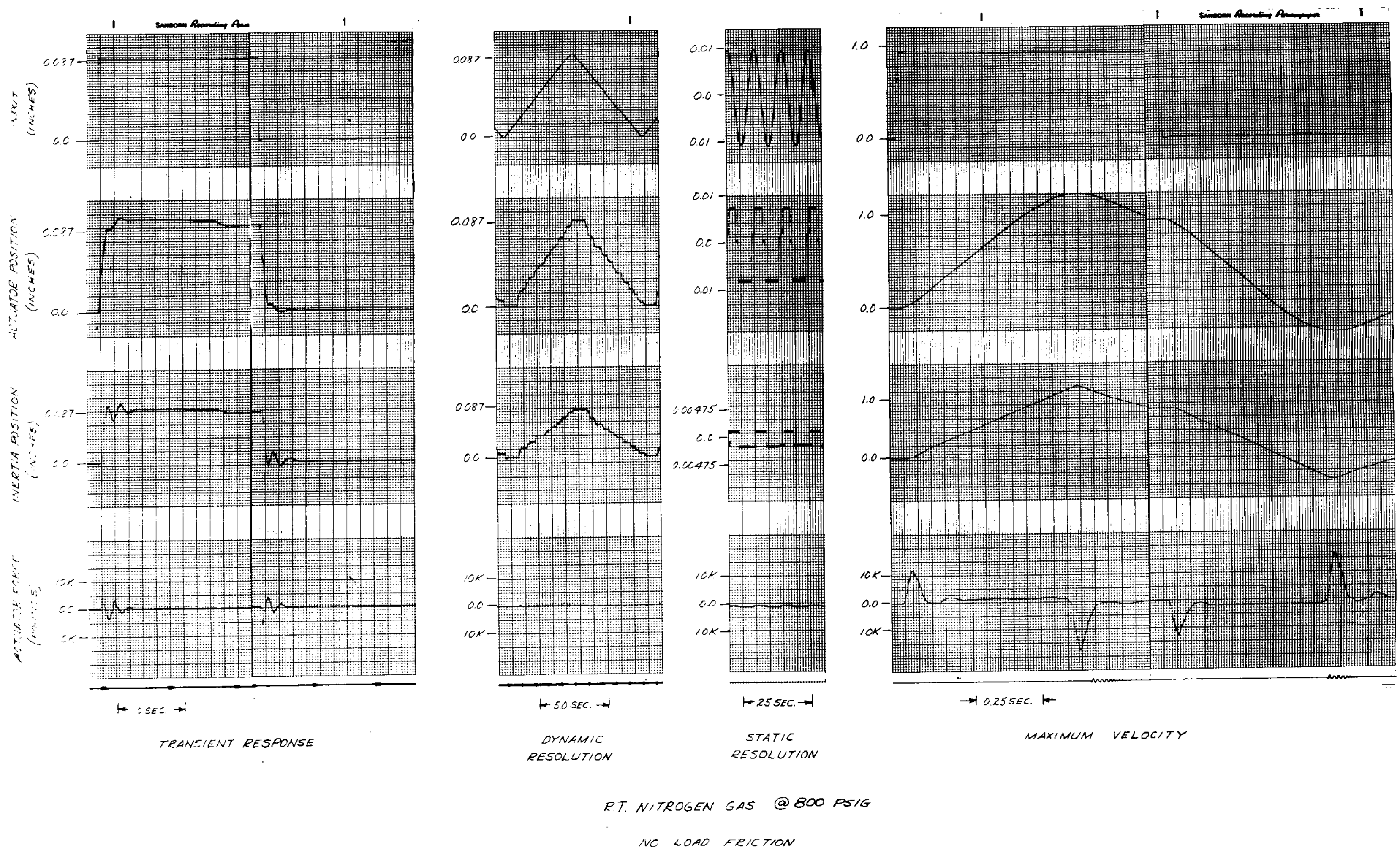
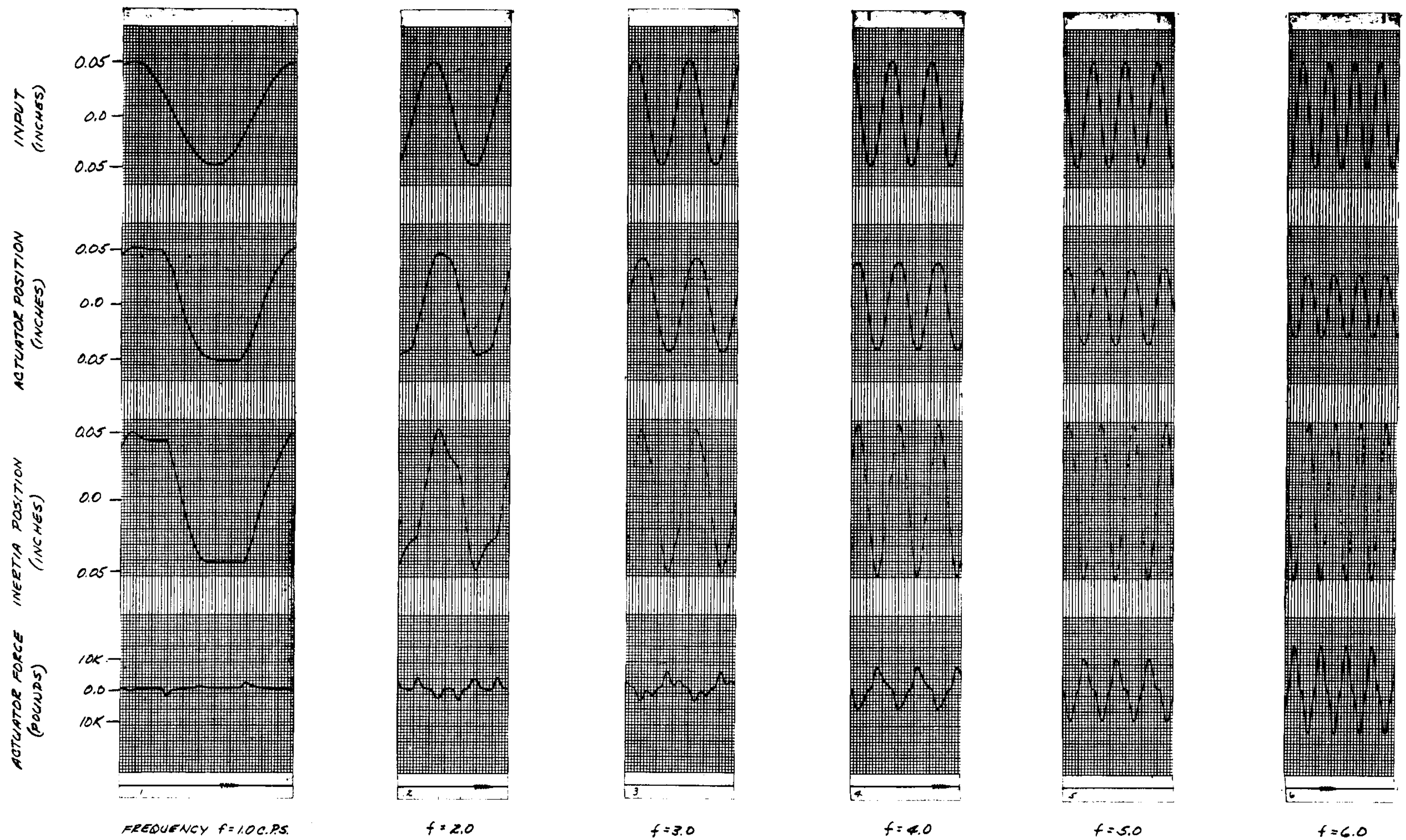


Figure 4-82. Actuator Closed Loop Performance.



R.T. NITROGEN GAS @ 800 PSIG

NO LOAD FRICTION

Figure 4-83. Actuator Closed Loop Frequency Response.

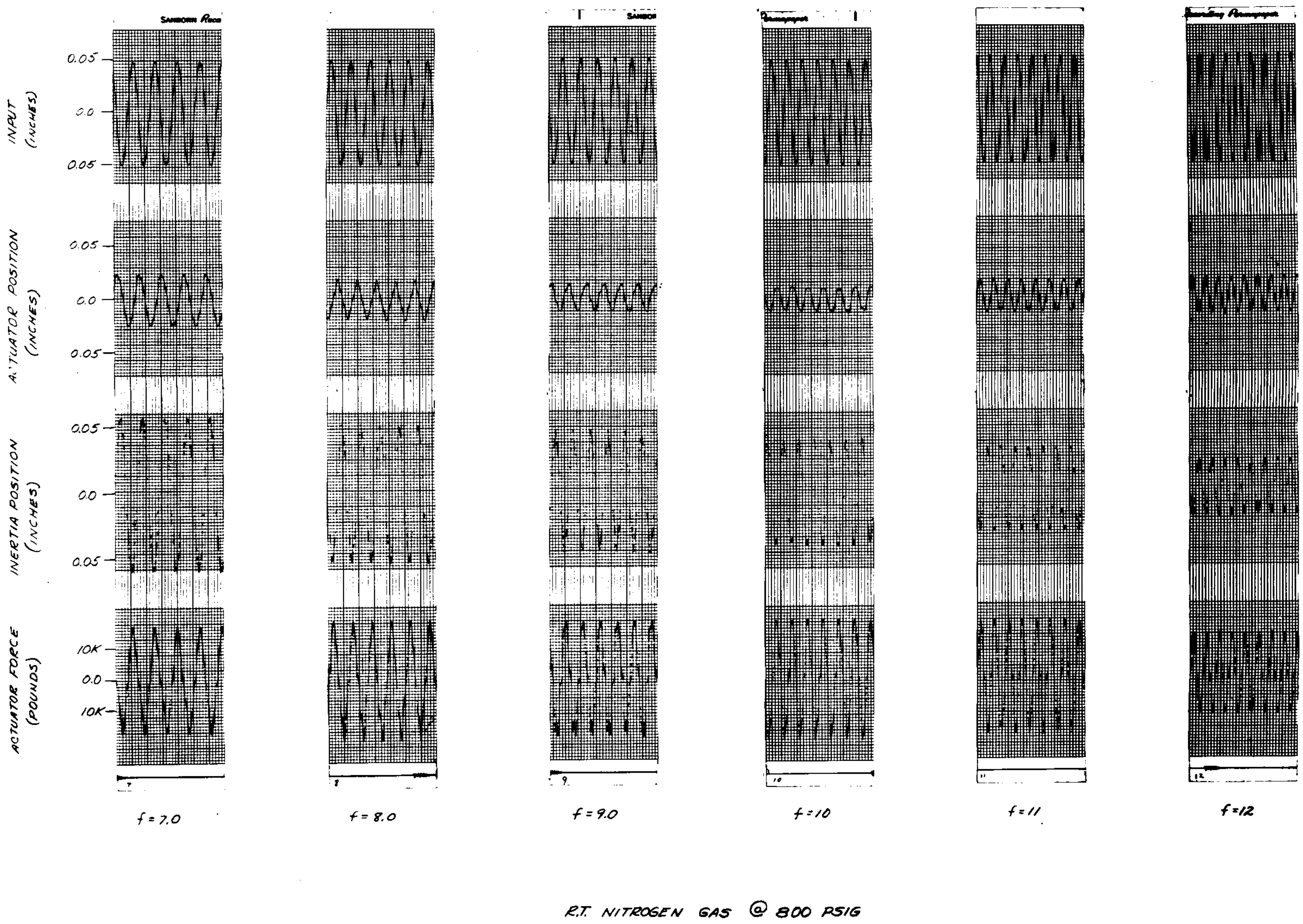


Figure 4-84. Actuator Closed Loop Frequency Response.

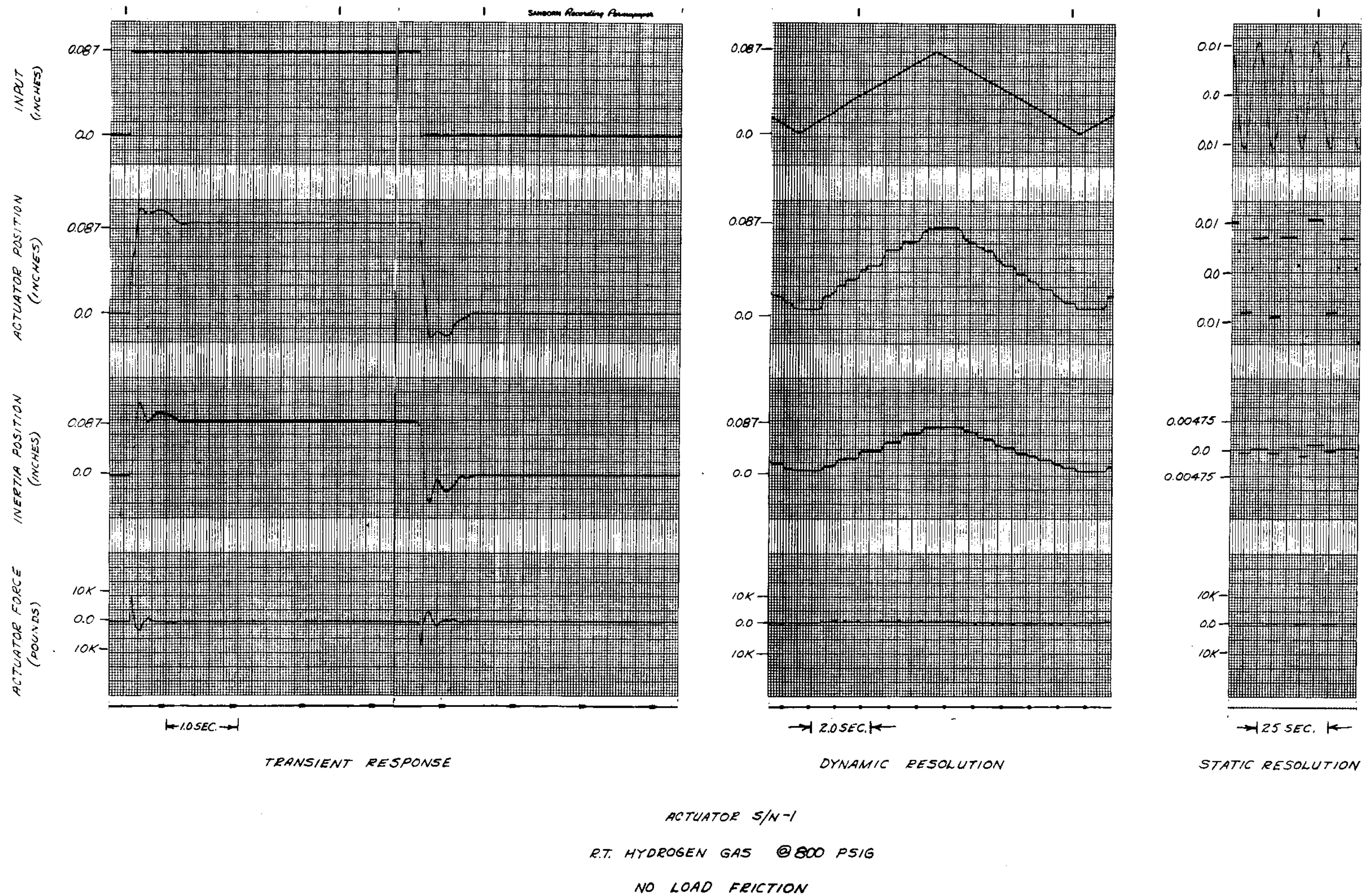


Figure 4-85. Actuator Closed Loop Performance.

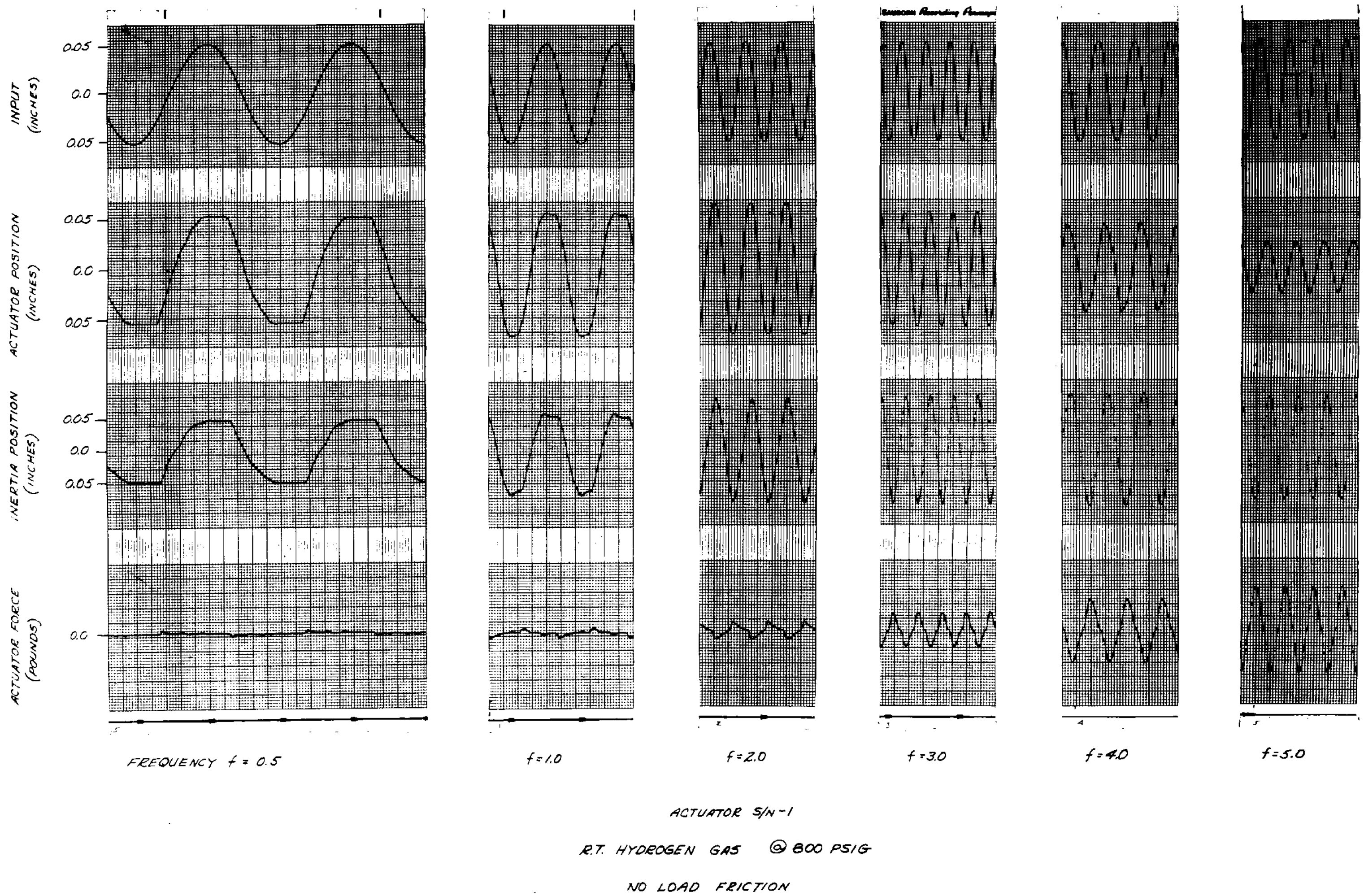


Figure 4-86. Actuator Closed Loop Frequency Response.

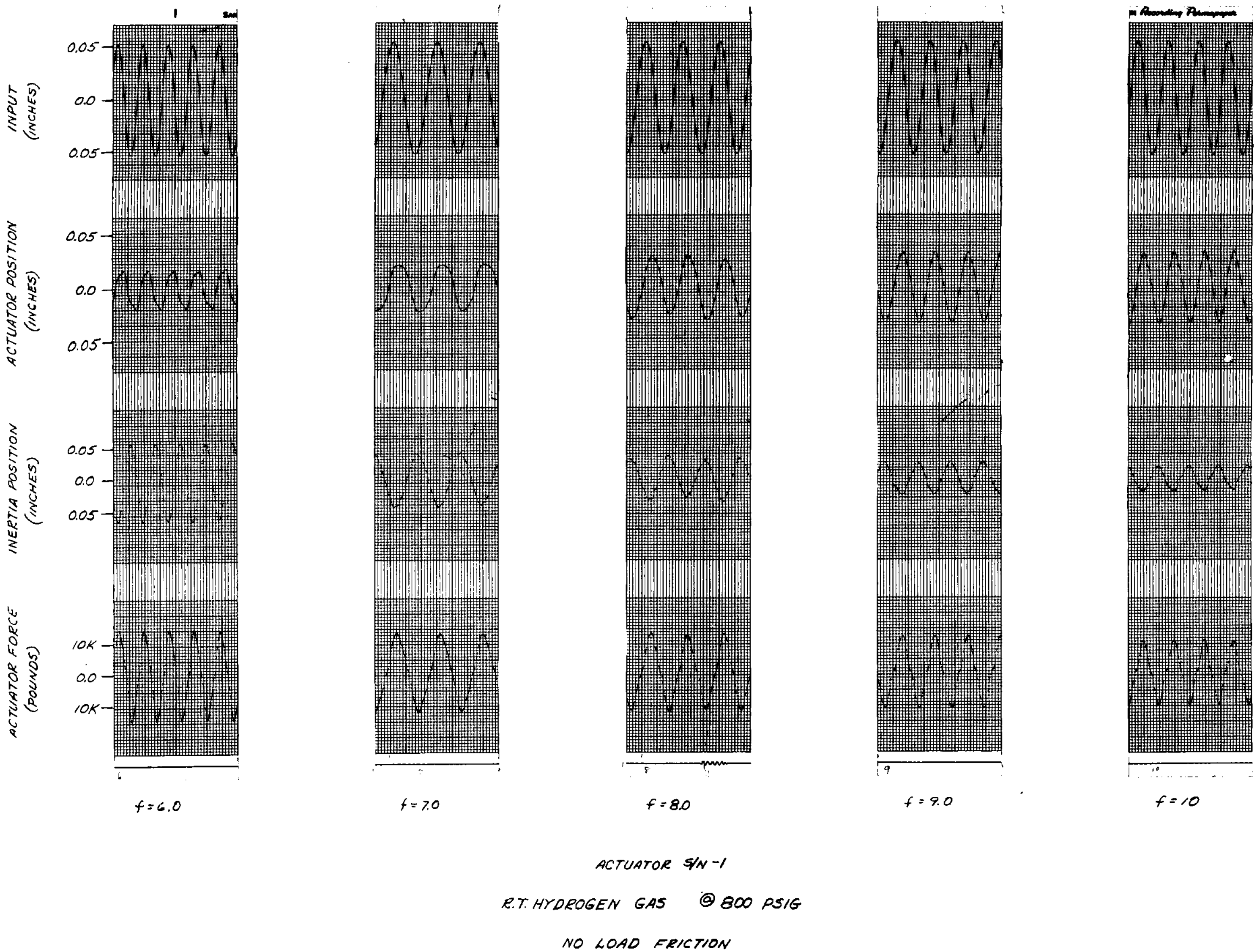


Figure 4-87. Actuator Closed Loop Frequency Response.

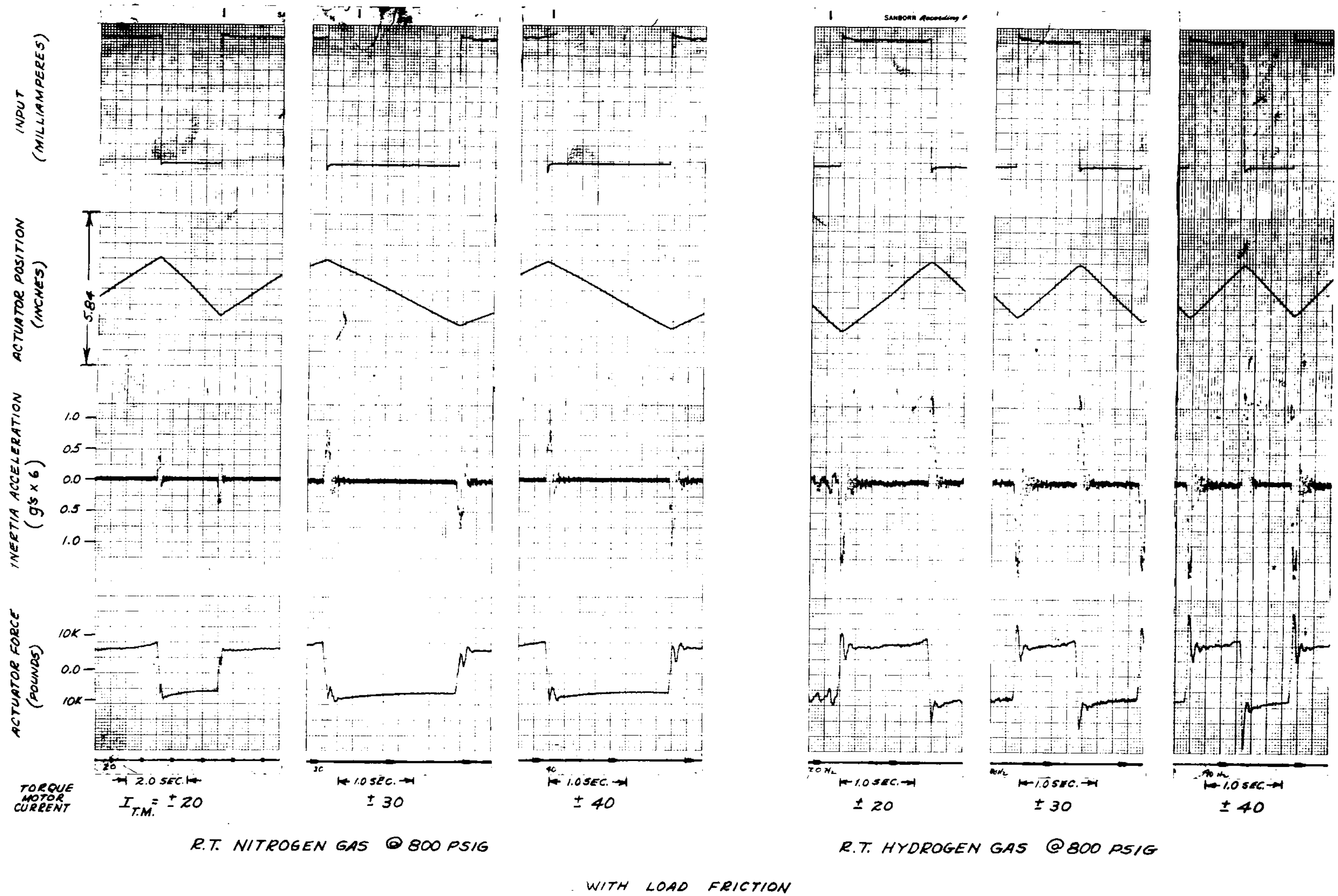


Figure 4-88. Load Inertia Acceleration.

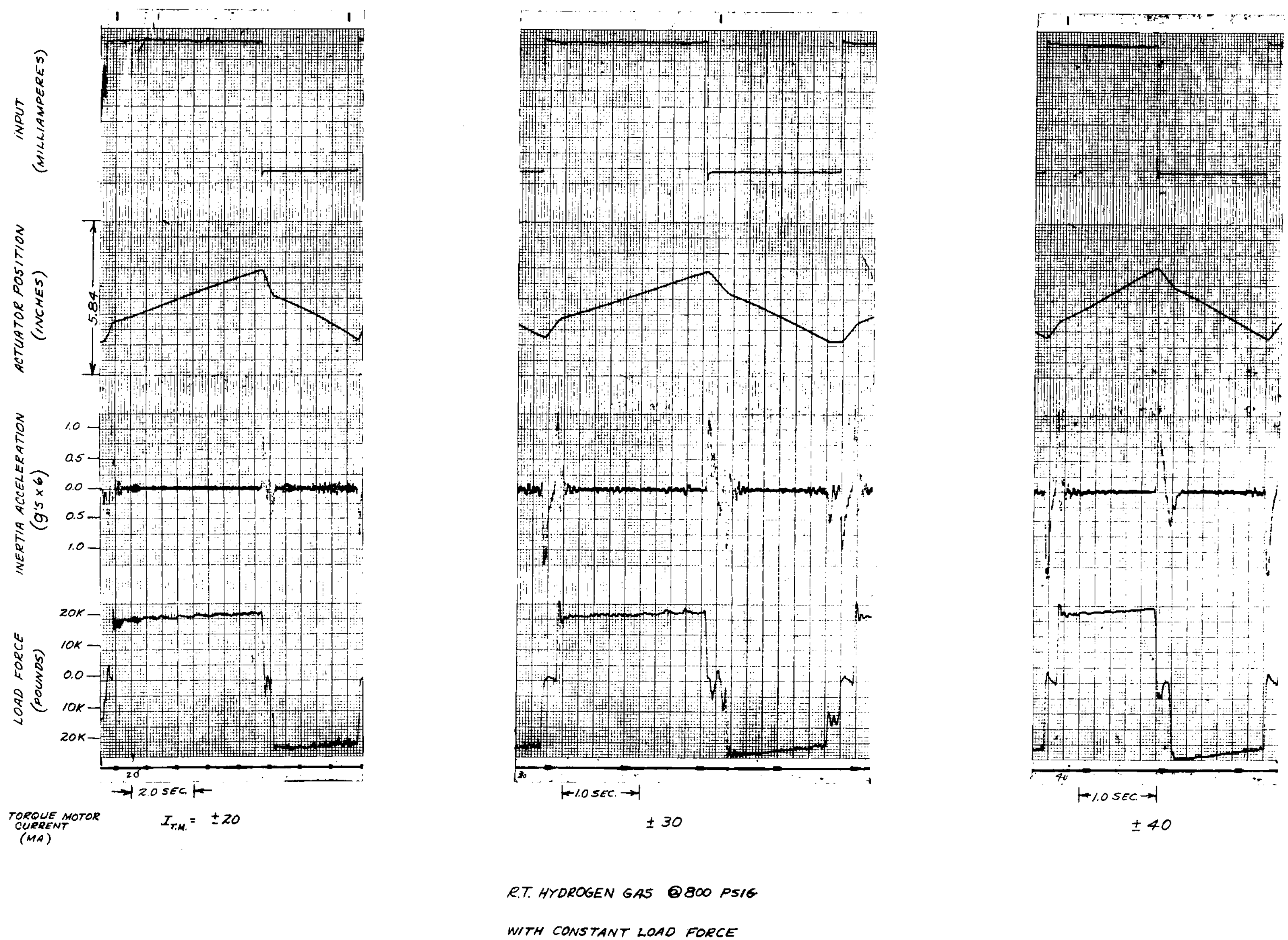


Figure 4-89. Actuator Velocity.

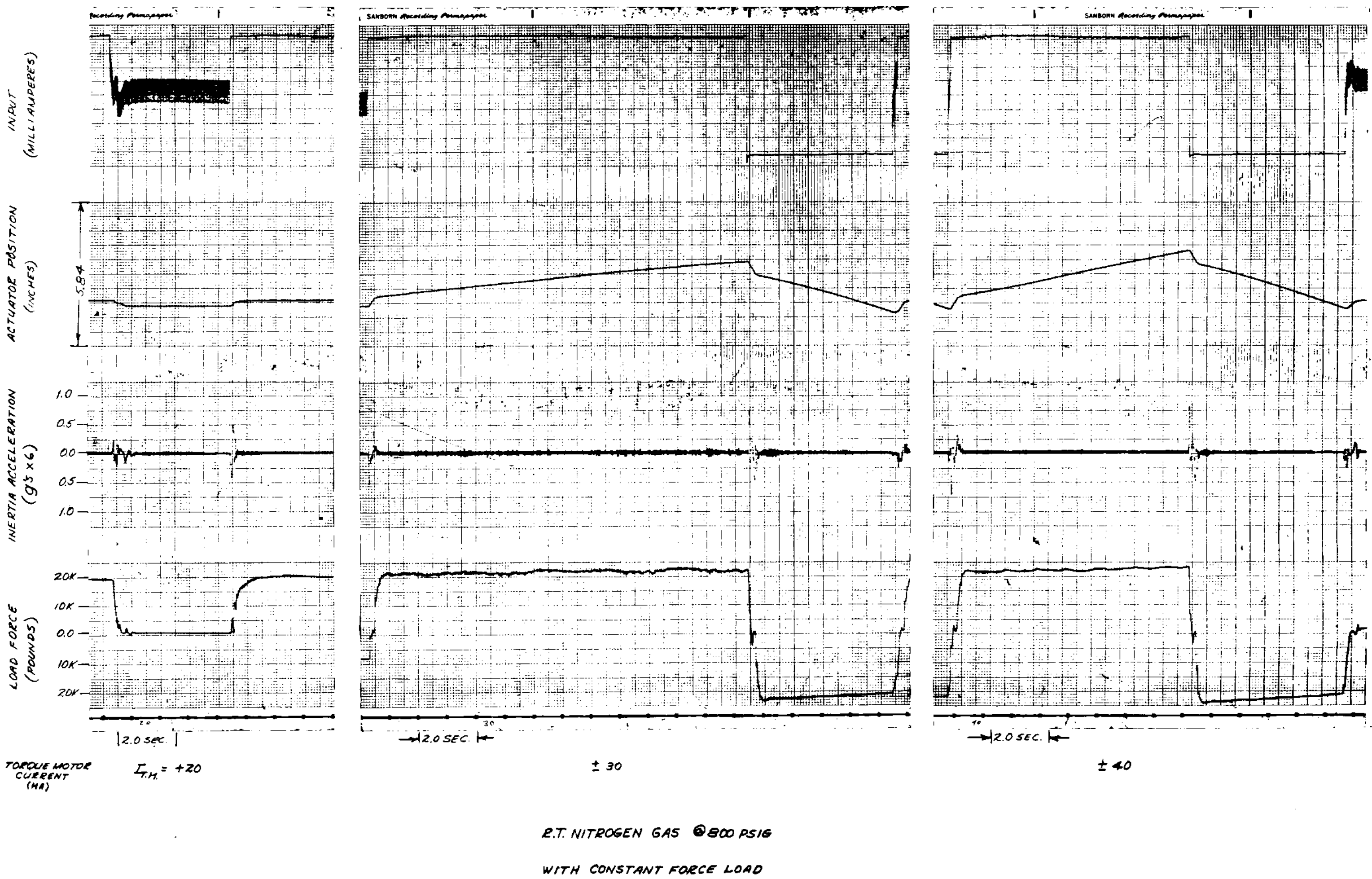


Figure 4-90. Actuator Velocity.

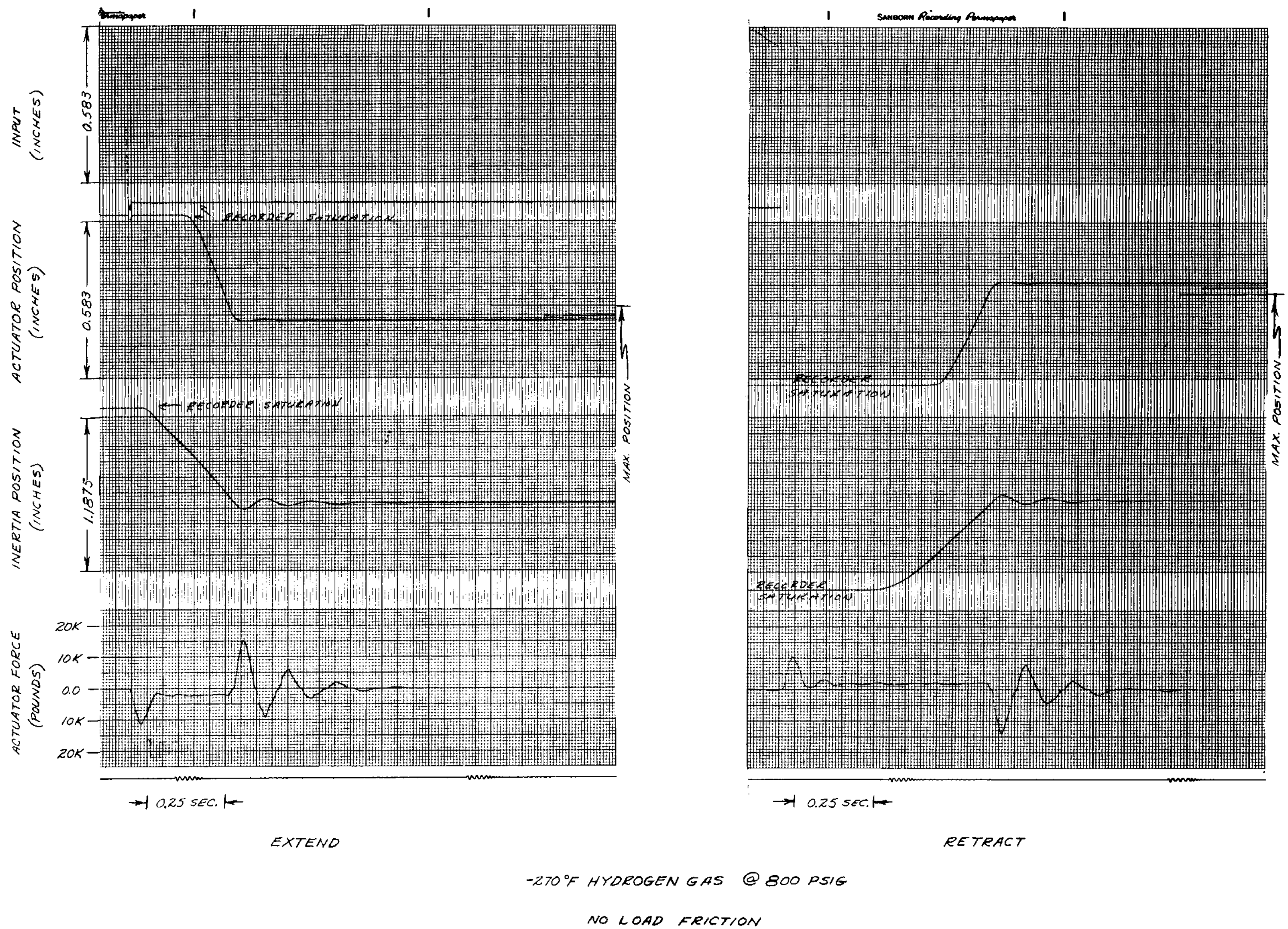


Figure 4-91. Snubbing.

A plot of the dynamics of the load fixture with no-load friction for the specified inertia and structural spring rate was not included, because a damping ratio was not defined and with a damping ratio of zero the load fixture would not be realistic.

The original design of the load fixture had provisions for mounting the actuator between the beam and a cantilever beam attached to the base of the fixture. Preliminary testing of this configuration indicated that its resonant frequency was approximately 4cps, and was primarily due to the spring rate of the beam supporting shaft. It was determined that if the beam supporting shaft was stiffened and the actuator was mounted between the beam and a hard stationary support, and the load inertia displacement was measured on the opposite side of the pivot, the load fixture would approximate the no-load friction configuration as will be shown.

The specified load with no-load friction has the configuration as shown in Figure 4-54.

Since the load inertia must necessarily damp out after it is disturbed, the damping ratio of the system was assumed to be small and in the order of 0.1.

The relation between the actuator displacement x_F and the inertia displacement x_B is:

$$\frac{x_B}{x_F}(S) = \frac{1}{\frac{72M}{K_1} S^2 + \frac{36D_T}{\ell^2 K_1} S + 1} \tag{1}$$

where

$$72M = 116.67 \text{ lbs. sec}^2/\text{in.}$$

$$K_1 = 391{,}000 \text{ lbs/in.}$$

Preliminary measurements of the load fixture used for the actuator evaluation indicate it to have a predominate spring rate at the beam pivot caused by deflection of the supporting shaft and its bearings as shown in Figure 4-55.

It is again assumed that there is damping and that it appears both in translation and in rotation.

The relation between the force input F_2 and the displacement of the actuator x_B is:

$$\frac{x_B}{F_2}(S) = \frac{\ell^2}{36D_T}\left[\frac{\frac{74M}{K_2} S^2 + \frac{D_F}{K_2} S + 1}{S\left(\frac{2\ell^2 M}{D_T} S + 1\right)\left(\frac{2M}{K_2} S^2 + \frac{D_F}{K_2} S + 1\right)}\right] \tag{2}$$

and the relation between the force input F_2 and the load inertia instrumentation potentiometer displacement x_C is

$$\frac{x_C}{F_2}(S) = \frac{\ell^2}{18D_T} \frac{-\frac{34M}{K_2}S^2 + \frac{D_F}{K_2}S + 1}{S\left(\frac{2\ell^2 M}{D_T}S + 1\right)\left(\frac{2M}{K_2}S^2 + \frac{D_F}{K_2}S + 1\right)} \tag{3}$$

combining equations (2) and (3), the relation between the load inertia instrumentation potentiometer displacement x_C and the actuator displacement x_B is

$$\frac{x_C}{x_B}(S) = \frac{2\left(-\frac{34M}{K_2}S^2 + \frac{D_F}{K_2}S + 1\right)}{\left(\frac{74M}{K_2}S^2 + \frac{D_F}{K_2}S + 1\right)} \tag{4}$$

The frequency response of x_C to x_B of the test load fixture which can be obtained from the test data shown in Figures 4-56 and 4-57 indicates that resonance occurs at 7.2cps, therefore

$$f_N = \frac{1}{2\pi}\sqrt{\frac{K_2}{74M}} \text{ cps} \tag{5}$$

where

$$M = 1.62 \frac{\text{lb. sec}^2}{\text{in.}}$$

$$f_N = 7.2 \text{ cps}$$

from which

$$\begin{aligned} K_2 &= (2\pi f_N)^2 (74M) \\ &= 4\pi^2 (7.2)^2 (74)(1.62) \\ &= 245{,}000 \frac{\text{lb}}{\text{in}} \end{aligned} \tag{6}$$

The load fixture frequency response indicates a damping ratio of approximately 0.18, therefore

$$\xi = \frac{D_F}{2\sqrt{74MK_2}} \tag{7}$$

from which

$$\frac{D_F}{K_2} = 2\xi \sqrt{\frac{74M}{K_2}}$$

$$= 2\ (0.18) \sqrt{\frac{74\ (1.62)}{24.5 \times 10^4}}$$

$$= 0.00786 \text{ sec.}$$

Substituting the coefficients into Equation (4) results in the following:

$$\frac{x_C}{x_B}\ (S) = 2\ \frac{-\ 2.25 \times 10^{-4}\ S^2 + 78.6 \times 10^{-4}\ S + 1}{4.89 \times 10^{-4}\ S^2 + 78.6 \times 10^{-4}\ S + 1} \tag{8}$$

Figure 4-56 shows a comparison of the Bode plots of the above equation and the actual load fixture with no-load friction. The coefficient 2 can be ignored because it was eliminated during the load fixture calibration.

Since all devices which have structural bending must necessarily consume energy, the damping of the load fixture is unavoidable. Assuming the actual engine to have a damping ratio but rather small, the load fixture as used is a fair approximation. The resonant frequency is 2cps below that desired, but the control system frequency response of command input to load inertia displacement with -270° F hydrogen gas was only attenuated minus 1.2db at 8cps. Had the load fixture resonance occurred at 9.2cps, the frequency response would have improved between 8 and 10cps.

When load friction is added to the load defined in the specification, the inertia should not move until the actuator compresses the structural spring an amount corresponding to the friction added. If 7300 pounds of friction force is used, the actuator should move 0.0186 inches before the inertia moves. At low forcing frequencies, the amplitude ratio of load inertia position to a command input of 0.05 inches would be minus 4db.

Since the load friction force is applied at point "D" on the load fixture used, and the load inertia position is measured at point "C", the beam will pivot about point "D" resulting in a displacement of point "C", even though the friction has not been overcome. When the torque supplied by the actuator is equal to the friction torque, the beam will rotate about the supporting shaft at point "A". The maximum displacement indicated by the load inertia position instrumentation potentiometer after the beam rotates about point "A" will be reduced by an amount corresponding to the displacement of point "C" prior to this rotation. Not recognizing this when evaluating the test data, one could mistakenly conclude that the resolution and stiffness of the actuator has deteriorated.

The frequency response test data for -270° F hydrogen gas and with load friction shown plotted in Figure 4-58 indicates at a frequency of 0.5cps the amplitude ratio of load inertia position to command input to be minus 7.15db with a gradual rise to minus 3.6db at 9cps. Reference to the load fixture frequency response plots shown in Figures 4-57 indicates that the load fixture as used has a low frequency attenuation of 1.6db more than it should. The low frequency amplitude ratio of the control system should therefore be minus 5.55 rather than minus 7.15 as shown. The difference between the low frequency amplitude ratio (-4db) of the analog computer plot of the specified load fixture dynamics and the adjusted test amplitude ratio (-5.55db) would be minus 1.55db and can be attributed to the control system stiffness and the motor friction characteristics.

As can be seen, the load fixture dynamics vary quite radically from the desired beyond 4cps. Rather than dropping off, the amplitude ratio increased with frequency to 7.2db at 9cps. This would account for the gradual rise in the control system frequency response.

Of interest is the low frequency phase shift shown in the plot of the analog computer simulation of the specified load. The plot shows a phase shift in the order of 30 to 35 degrees. This would indicate that to obtain no more than 20 degrees phase shift at 1cps with an infinitely stiff actuator would require additional low frequency phase lead compensation.

4.3.3 Assembly and Operational Difficulties Experienced

4.3.3.1 Thrust Bearing Failure

After completion of cold testing of S/N 2 actuator and during some room temperature slew velocity tests, the actuator was jammed. On disassembly it was found that bearing retaining nut 2775640 had backed off after failing the safety wire. This permitted the ball nut gear to slide back against the main housing, cutting off the safety wire to retaining nut 2775639. On retraction, the gear slid forward again and jammed nut 2775640 against cover 2775616. At this time the actuator would not operate. Further investigation indicated that the matched set of four-point contact bearings had been installed incorrectly so that the one bearing (probably the center bearing) was taking all the load instead of it being shared by three bearings. A bearing jammed and the gear was turned within the inner race applying a sufficient torque to retaining nut 2775640 to fail the lockwire. The nut was then free to move and the gear was not secured to the bearings. The center bearing was disassembled but indicated only a build-up of SP-1 ball retained material on the races. The races had not failed. On reassembly with a new set of bearings, the retaining nut was locked to the gear by a radial pin installed at assembly.

4.3.3.2 Loss of Balls from Ball Spline Assembly

During full stroke operation of S/N 2 actuator at low loads, balls were lost out of the head end row of the ball retainer on two separate occasions. At the time this difficulty was discovered there was not sufficient time to perform any testing to determine the cause of the loss of balls. The last row of balls (6 balls out of 30) were removed from S/N 2 actuator after determining that the ball loads were not excessive.

The loss of balls may have been due to a binding condition in the ball retainer, excessive centering spring force, and/or inertia effects of the ball retainer. Further investigation of this difficulty including all forces involved vs. driving force of balls at various resistive torques will be necessary to determine the cause.

4.3.3.3 Housing Spanner Nuts

The large spanner nuts, P/N 2775617 and 2775618, used to attach cover 2775616 and bracket eye assembly 2775665 to the housing were originally designed to eliminate parts and provide a very stiff housing in tension. It was found, however, that the friction of such a large thread and the friction of the nut shoulder against the part secured required such a torque that it was not possible to properly mash the metallic "O" rings used to seal at each end of the main housing. It was necessary to install the actuator in the test fixture and apply load with the fixture load cylinder in order to properly preload these spanner nuts before torquing. No change could be made on these units.

4.3.3.4 Potentiometer Support

Vibration tests on the original support for the connector end of the potentiometer indicated insufficient support. See section 4.2.6 of this report. In order to better support this potentiometer,nut 2775643 was redesigned and compression spring 2775828 was added. It was also necessary to make Change B in potentiometer assembly 2775565. S/N 2 actuator was equipped with these new parts, but S/N 1 was not corrected because of the delivery requirement.

4.3.3.5 Rotation of Cover, P/N 2775616

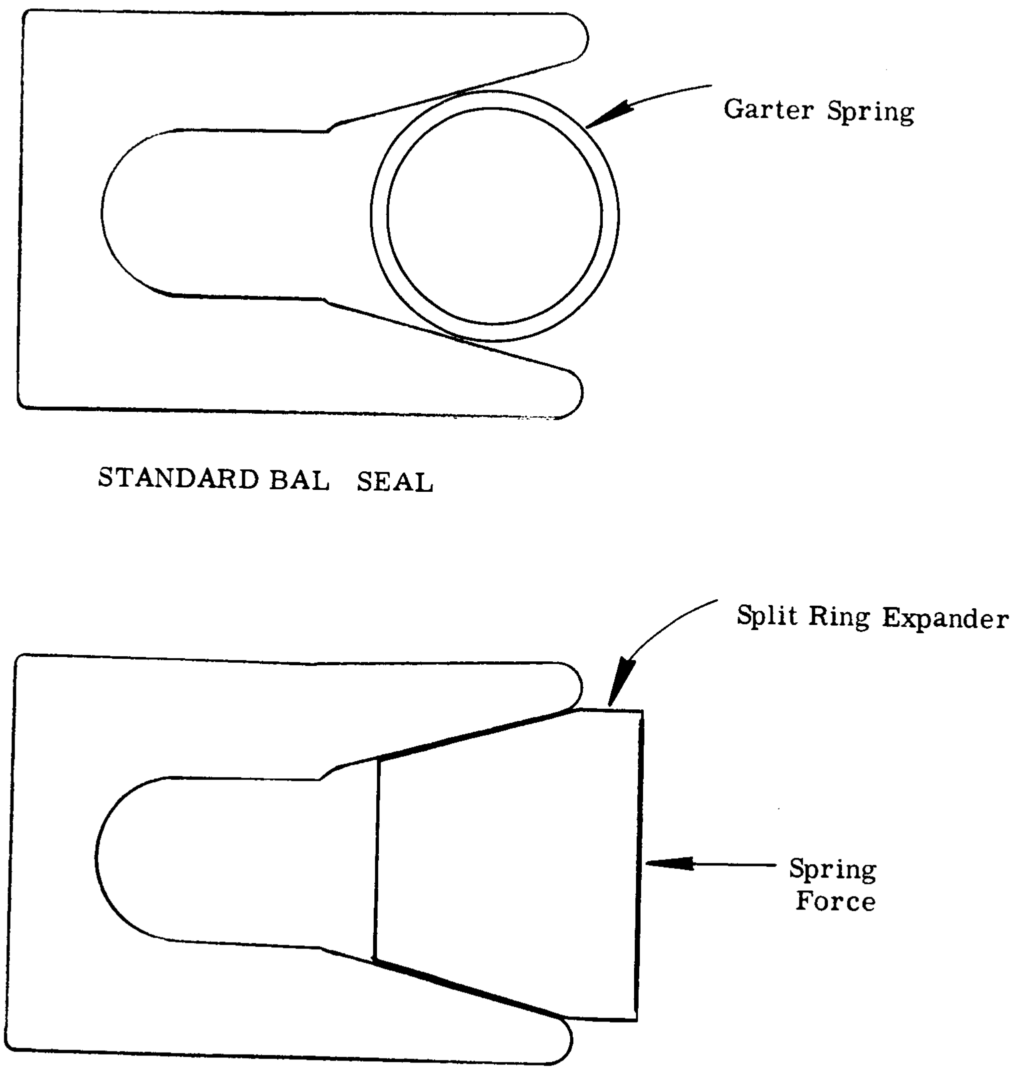
It was found on S/N 1 actuator that the cover, P/N 2775616, had rotated slightly when the unit was loaded on the test rig to mash the metallic "O" ring seal and spanner nut 2775617 was torqued. This provided some difficulty in aligning the travel indicator. To correct this on S/N 2 actuator, a change was incorporated on cover 2775616 so that it would be doweled to the main housing thereby preventing rotation of the cover.

4.4 DYNAMIC SEAL AND LINEAR MOTION BEARING DEVELOPMENT

The purpose of these tests was to evaluate the Bal seal, P/N 1005-238, for use in the NV-B1 actuator to seal hydrogen gas between the output shaft and the main housing and to evaluate Rulon tape for use as the support bearing between the output shaft and main housing.

4.4.1 Description of Parts Tested

The Bal seal, P/N 1005-238, is a small cross section cup-type diametral seal utilizing a garter spring to maintain contact between the seal and inner and outer diameters. Seals manufactured from Teflon and Kel F were tested. Figure 4-92 shows the seal cross section.



STANDARD BAL SEAL

BAL SEAL WITH SOLID EXPANDER

Figure 4-92. Bal Seal With Solid Expander.

The bearing material tested was Rulon Type A 0.033 inch thick, a modified Teflon manufactured by the Dixon Corporation. The projected bearing area is 1.87 square inches. Figure 4-93 shows the bearing and installation dimensions.

The test fixture used was NPX-102-128. The main parts include a cylinder and a shaft carrying two pistons. A Bal seal and a Rulon bearing were fitted to each piston. The piston and cylinder material was AMS 5610 R_c 26-33. The surface finish of the cylinder bore was 4 to 6 micro inches.

The seal groove and cylinder dimensions used were as recommended in the Bal Seal Catalogue; both one-piece and two-piece groove constructions were used. The two-piece construction allowed variation of the garter spring compression by the addition of shims behind the seal.

4.4.2 Background

4.4.2.1 Dynamic Seal

Numerous seal companies were contacted by letter in an effort to obtain a dynamic seal for the NV-B1 output shaft. None could offer a leak-tight seal for the -250° F temperature requirement except the Bal Seal Engineering Company which manufacture a standard seal of suitable dimensions.

The acceptable leakage chosen for this seal was 3.75×10^{-2} scc/sec. using hydrogen gas at -250° F and 50 psia. This number was calculated on the following premise which was obtained on the NERVA program. The maximum seal leakage is 10^{-2} scc/sec/inch of seal diameter under the normal operating range of temperature and pressure. The required operating conditions for this seal are:

Hydrogen gas at 0-50 psia and -250° F to 150° F

The nominal seal diameter is 3.75 inches.

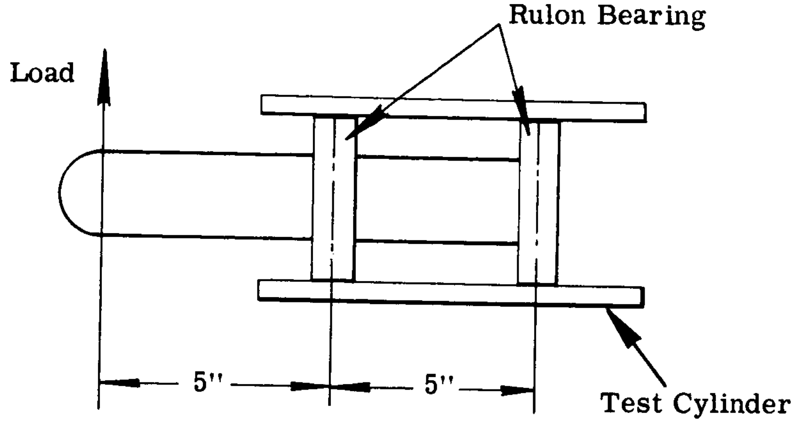
4.4.2.2 Linear Motion Bearing

The bearing tests were made in conjunction with the seal tests, because the bearing and seal share a common rubbing surface in the main housing and common test fixturing was employed.

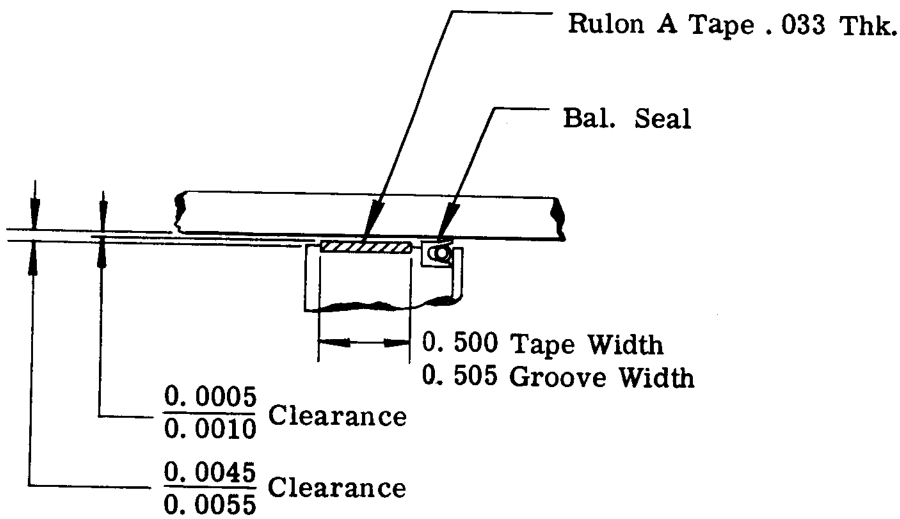
The purpose of the bearing is to enable the actuator to withstand the 5000-pound side load as specified for the actuator.

4.4.3 Test Results

4.4.3.1 Static Leak Checks



METHOD OF LOADING BEARING



LINEAR MOTION BEARING INSTALLATION

Figure 4-93. Linear Motion Bearing Installation.

4.4.3.1.1 Room Temperature Tests

The Teflon seal showed no leakage when pressurized from 0-300 psig with nitrogen gas. Both one-piece and two-piece piston groove construction was used. With the one-piece groove construction, it was extremely difficult to assemble and impossible to remove the seal without damage.

The Kel F seal was tested under similar conditions and the leakage rate exceeded the maximum by a factor of 200.

4.4.3.1.2 Preliminary Cryogenic Leak Checks

Preliminary checks showed that the seal would develop a sudden rise in leakage as the temperature was reduced. The gas pressure was increased to 150 psig to determine if the greater sealing force developed would help the situation. The result was a more dramatic increase in the leakage.

4.4.3.1.3 Reduced Seal Groove Width

Shim stock was added to the seal groove to provide a further .006 inch compression of the seal garter spring and a more comprehensive test setup including thermocouples was used. A gas leakage test produced the following results:

Temperature - °F	Leakage
+75	0
-75	0
-114	0
-170	0
-190	Leakage
-201	915 cc/min.
-226	1820 cc/min.
-247	28300 cc/min.

4.4.3.1.4 Seal Reaction to Cryogenic Temperatures

The piston and seal assembly was removed from the cylinder and cooled to approximately -250° F. Contraction of the seal outer lip was observed to compress the garter spring such that the seal outer diameter was less than the cylinder diameter.

4.4.3.1.5 Modified Bal Seal

The seal garter spring was replaced with a solid tapered section split expander ring. Wedging action of the seal to the piston and cylinder diameters was achieved by the installation of a helical compression spring under each of the six screws retaining the movable side of the seal groove. The compression springs were set to provide a total force on the piston ring of 60 pounds; a force of 21 pounds was required to move the piston assembly along the cylinder.

The leakage was 9 scc/min. with helium gas at 70°F and 10 psig. The seal temperature was then reduced to -275°F and the leakage remained constant at 9 scc/min. It should be noted that the cylinder was scored from previous testing and thus it was impossible to achieve a complete seal.

Sufficient force was not immediately available to give linear motion to the piston, but a torque of approximately 800 inch-pounds was required to rotate the piston assembly at a temperature of -200°F with the seal open to atmosphere.

Assuming the coefficient of friction for linear motion is the same as the rotational motion, the equivalent force is 427 pounds.

The leakage rate was increased to 12 scc/min. at -200°F after rotation of the seal.

4.4.3.2 Seal Life

4.4.3.2.1 Teflon Seal

The teflon seal was too badly worn to continue the life test without risking ruining the cylinder finish after 6000 cycles of ±0.75 inch at 20cpm with nitrogen gas at 10 psig applied to the seal.

4.4.3.2.2 Kel F Seal

The Kel F seal was cycled at the same conditions as the Teflon seal and after 174,000 cycles showed negligible wear.

4.4.3.3 Bearing Test Results

Applied Load - Lbs.	Force To Move Out Lbs.	Force To Move In Lbs.	Effective Load On No. 1 Bearing Lbs.
320	73.9	83	640
643	151	144	1286
1288	302	354	3576
1930	457	475	3860
2570	604	600	5140

Projected area of bearing = 1.87 sq. in.

Maximum compressive stress applied 5140/1.87 = 2740 psi

Maximum compression measured 7% (measured at contact center) average coefficient of friction = .08

4.4.4 Conclusions

4.4.4.1 Dynamic Seal

The Bal seal will provide an effective seal at temperatures as low as -170° F, but below this value the contraction forces within the seal cause compression of the garter spring rendering the seal ineffective. Replacing the garter spring with a split, solid, spring loaded expander ring will maintain a seal at -250° F, but with an attendant large increase in friction which may cause a high wear rate.

The seal life is inadequate when operating a Teflon seal against an AMS 5610 stainless steel cylinder. Seal life may be increased by use of hard chrome plate and a molybdenum disulphide dry film lubricant.

4.4.4.2 Linear Motion Bearing

The Rulon bearing as designed will handle the 5000-pound side load specified for the NV-B1 actuator. The life of the bearing was not determined, but no difficulty is anticipated in achieving the 100 full cycles specified.

4.4.5 Recommendations

4.4.5.1 Dynamic Seal

The Bal seal should be used in the NV-B1 actuator for the following reasons:

a. No alternate seal is readily available; thus a seal development program is required to obtain a satisfactory seal, followed by modification of the actuator to accommodate the new seal.

b. Seal leakage is a hazard on these first two actuators only when tested with hydrogen gas. Since the available supply of hydrogen gas will limit hydrogen running to tests of short duration, it is unlikely that the operating temperature of the seal will exceed its working range.

c. Tests of longer duration will be performed with air or nitrogen gas; the properties of which preclude operation outside the temperature range of the seal.

d. The seal life will be lengthened by the use of hard chrome plate on the cylinder bore.

Seal wear should be monitored throughout the actuator testing to obtain data with regard to seal life when used in a hard chrome bore.

A seal development program should be initiated to obtain a seal suitable for a flight version of the NV-B1 actuator.

4.4.5.2 Linear Motion Bearing

The Rulon bearing should be incorporated in the NV-B1 actuator using the same dimensions as the test bearing.

The performance of the bearing should be monitored during the actuator testing to verify that the bearing is satisfactory, with emphasis on performance during the actuator side loading tests.

4.5 SNUBBER SPRING DEVELOPMENT TEST

4.5.1 Purpose

To determine the spring rate of the disc spring P/N 2775578 and to ensure that dynamic loading at -250° F will not cause structural failure.

4.5.2 Description of Parts Tested

This disc spring was designed for snubbing the Model NV-B1 actuator, four springs will be used in each actuator, 3 springs in series and one inactive.

The spring distorted during heat treatment and in order to obtain a flat contact surface with a minimum width of 0.030, the spring height was reduced from $\frac{.557}{.555}$ to .552 inch.

The surface finishes of the part did not meet the minimum requirement of 32 micro inches and were probably no better than 125 micro inches.

The final static test was made after the spring outer diameter was machined from a conical to a cylindrical form, the final diameter being 4.900 inches.

The maximum spring travel available is 0.040 inch; the maximum working travel is 0.033 inch or 82% of maximum available.

4.5.2.1 Test Equipment

4.5.2.1.1 Static Test

The static test was performed on an Olsen testing machine and the load was applied to the spring via a 3-inch diameter steel ball to ensure even loading.

4.5.2.1.2 Dynamic Test

Two methods were used for dynamic testing:

a. The first method consisted of a 50-pound steel cylinder dropped through a steel guide tube. The spring was resting on a 6" diameter x 4" thick cylinder which in turn was resting on the concrete floor.

b. The second test utilized a small drop test rig. The platten weight was 533 lbs. This was increased by resting seven 100-lb. lead bars on the lower I beam of the platten.

In both tests, spring travel was determined by measuring solder strips crushed by the lower moving edge of the spring.

4.5.3 Summary of Results

4.5.3.1 Static Test

The static test results are shown in Figures 4-94 and 4-95. The spring rate and hysteresis were 2.05×10^6 lb/in and 20% lb/lb respectively. After machining the outside diameter from conical to cylindrical form, the spring rate and hysteresis were 2.0×10^6 lb/in and 24% lb/lb respectively.

4.5.3.2 Dynamic Tests

4.5.3.2.1 First Test Rig

The first test failed to load the spring evenly and the resulting deflection measured at different points around the spring varied from 0 to 0.015 inch.

The impact load of 1210 in-lbs was applied to the spring three impacts at room temperature and ten impacts at approximately -300° F. No physical damage was evident at the completion of this test.

4.5.3.2.2 Second Test Rig

The second test comprised six impacts which were increased in energy from 1210 in-lbs to 11,400 in-lbs to achieve required 0.033 inch. The plot of spring travel vs. $\sqrt{\text{Load Energy}}$ is shown in Figure 4-96.

4.5.4 Conclusions

The static tests showed that the spring rate was 2.0×10^6 lb/in which was the design objective.

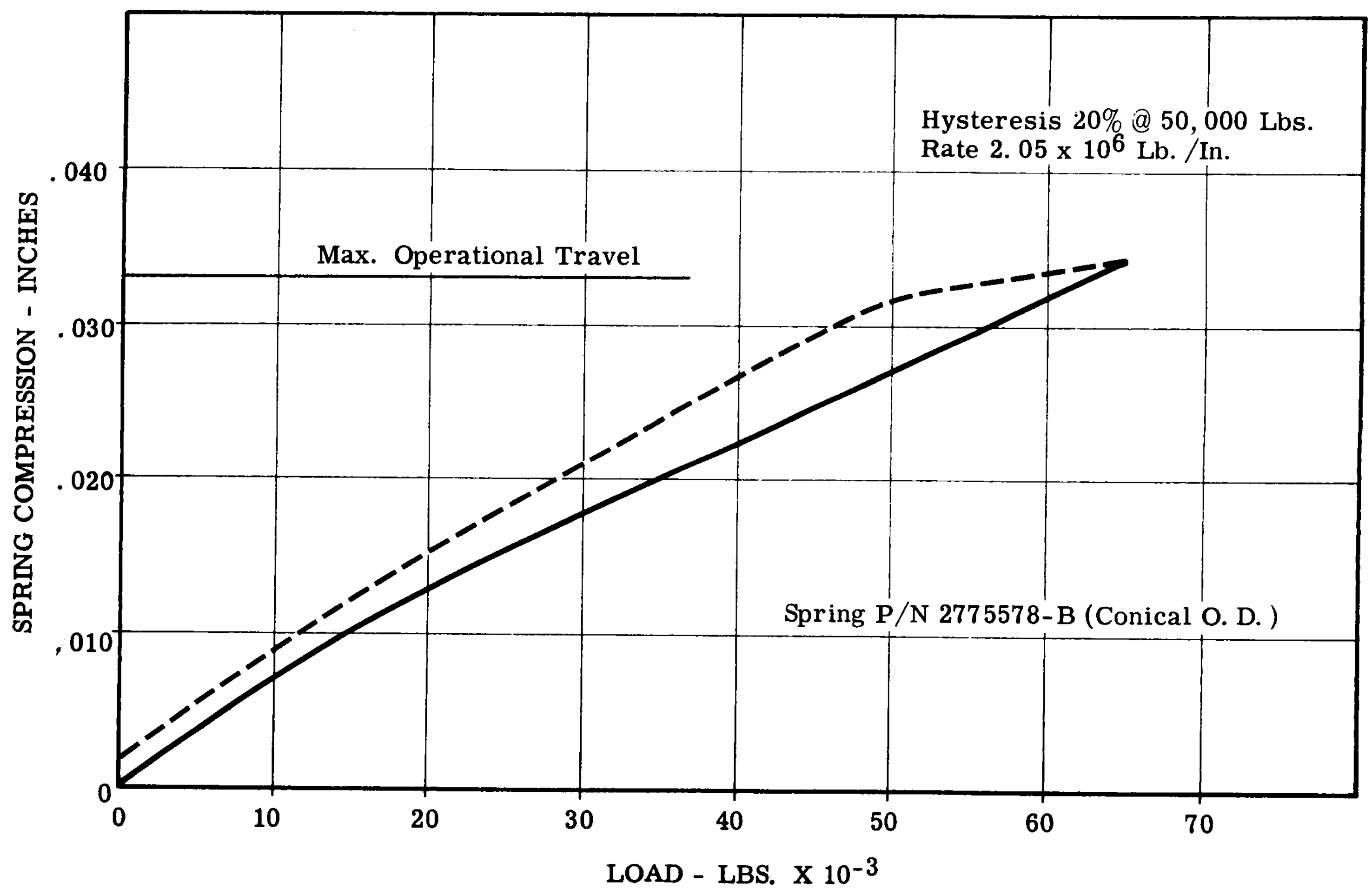


Figure 4-94. Disc Spring Characteristic Curve (Conical O.D.).

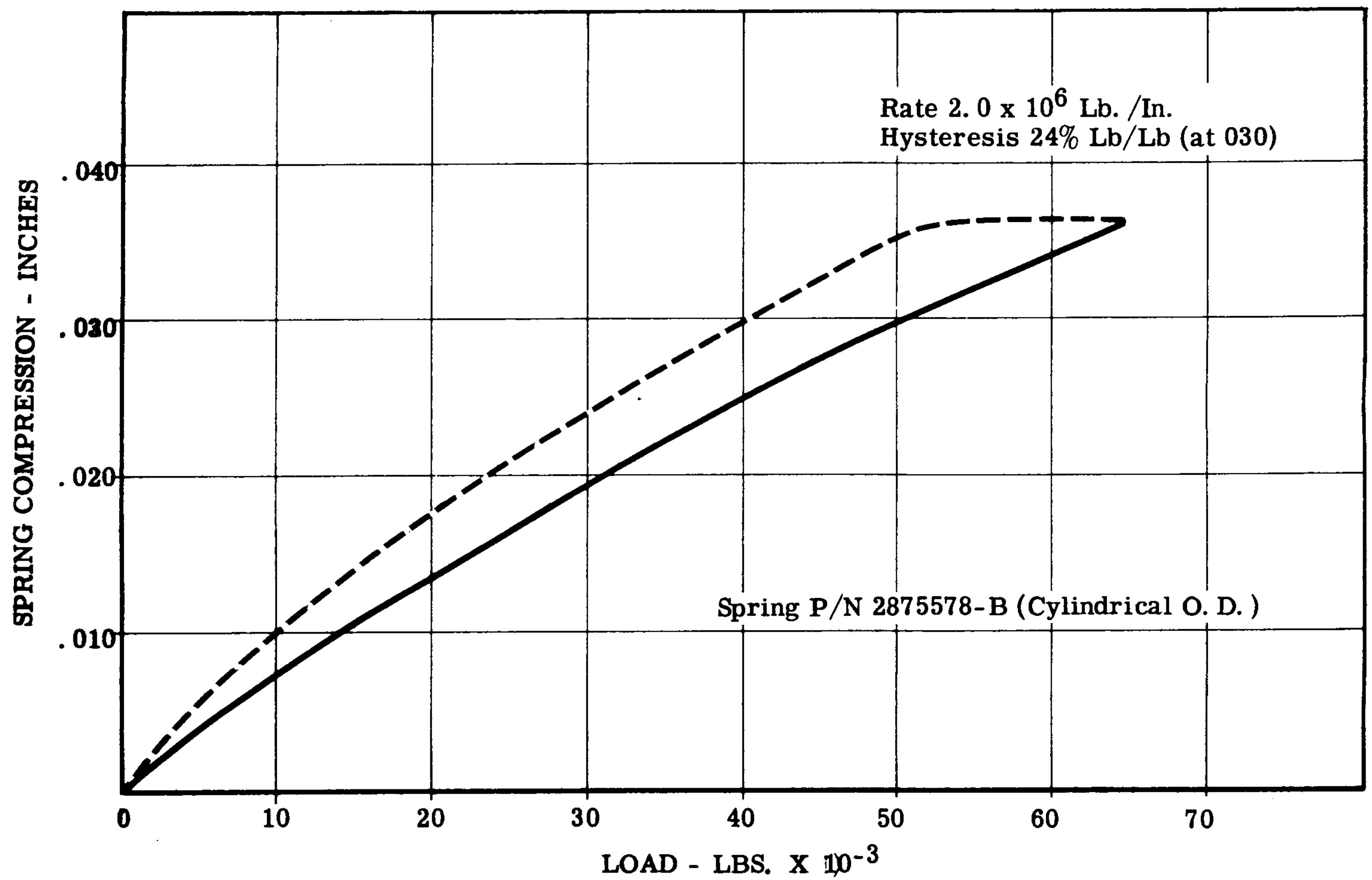


Figure 4-95. Disc Spring Characteristic Curve (Cylindrical O.D.).

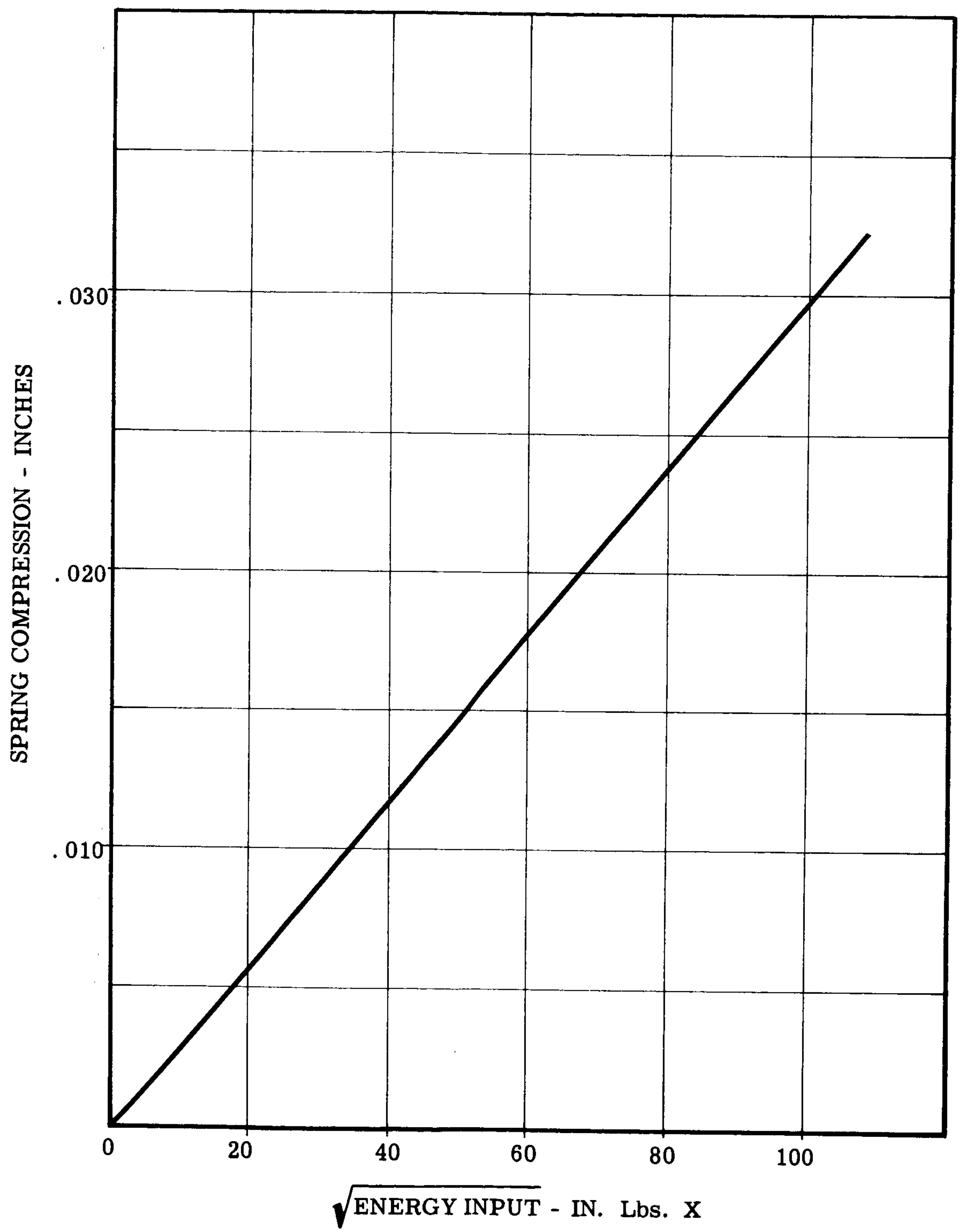


Figure 4-96. Spring Impact Load Characteristic.

The dynamic tests showed that the spring may be dynamically loaded through the designed working stroke of 0.033 inch without exceeding the elastic limit of the spring at room temperature.

The dynamic tests at -300° F were made only on the first test setup and the maximum spring travel achieved was approximately 0.017 inch. However, this test was repeated 10 times and since the minimum required operating temperature is -250° F it may be assumed that the spring will meet the environmental requirements.

Correlation between the theoretical and actual spring travel was not achieved, the energy input for 0.033 inch of travel was almost 10 times the anticipated value. Investigation of this situation has not revealed the problem area.

4.5.5 Recommendations

The spring thickness should be increased from .515/.513 to .532/.529 inch to absorb the increase in allowable maximum actuator output force, 34,800 lbs. to 42,000 lbs. agreed to after the test spring was designed.

4.6 ACTUATOR STIFFNESS

Two tests were performed on the NV-B1 actuators to determine the stiffness or structural spring rate of the actuator.

The first test was performed with the actuator (less the servo motor) mounted on the engine simulation fixture and with the sun gear of the planetary transmission locked to determine the overall mechanical spring rate of the actuator. The actuator was then loaded to 15,000 pounds and deflection was measured by the actuator potentiometer position change. The spring rate after overcoming initial friction forces was found to be 1.6×10^6 lbs/in. as shown in Figure 4-97.

The second test was performed with the complete closed loop actuator system mounted on the engine simulator fixture and loaded in the same manner as above using the potentiometer position indicator to measure deflection. A complete hysteresis loop of forces was applied. In compression the average spring rate was found to be 2.28×10^6 lbs/in. whereas in tension the average rate was 1.67×10^6 lbs/in. See Figure 4-98 for the closed loop stiffness curve.

4.7 VIBRATION OF COMPLETE UNIT

The S/N 2 actuator was vibrated through the complete spectrum in all three planes. The unit was not endurance vibrated for an hour in each plane since there was no desire to damage the actuator.

4.7.1 Method of Test

The actuator was mounted on a fixture which allowed the actuator to be vibrated through one axis perpendicular to the ball screw centerline. Reversal of the

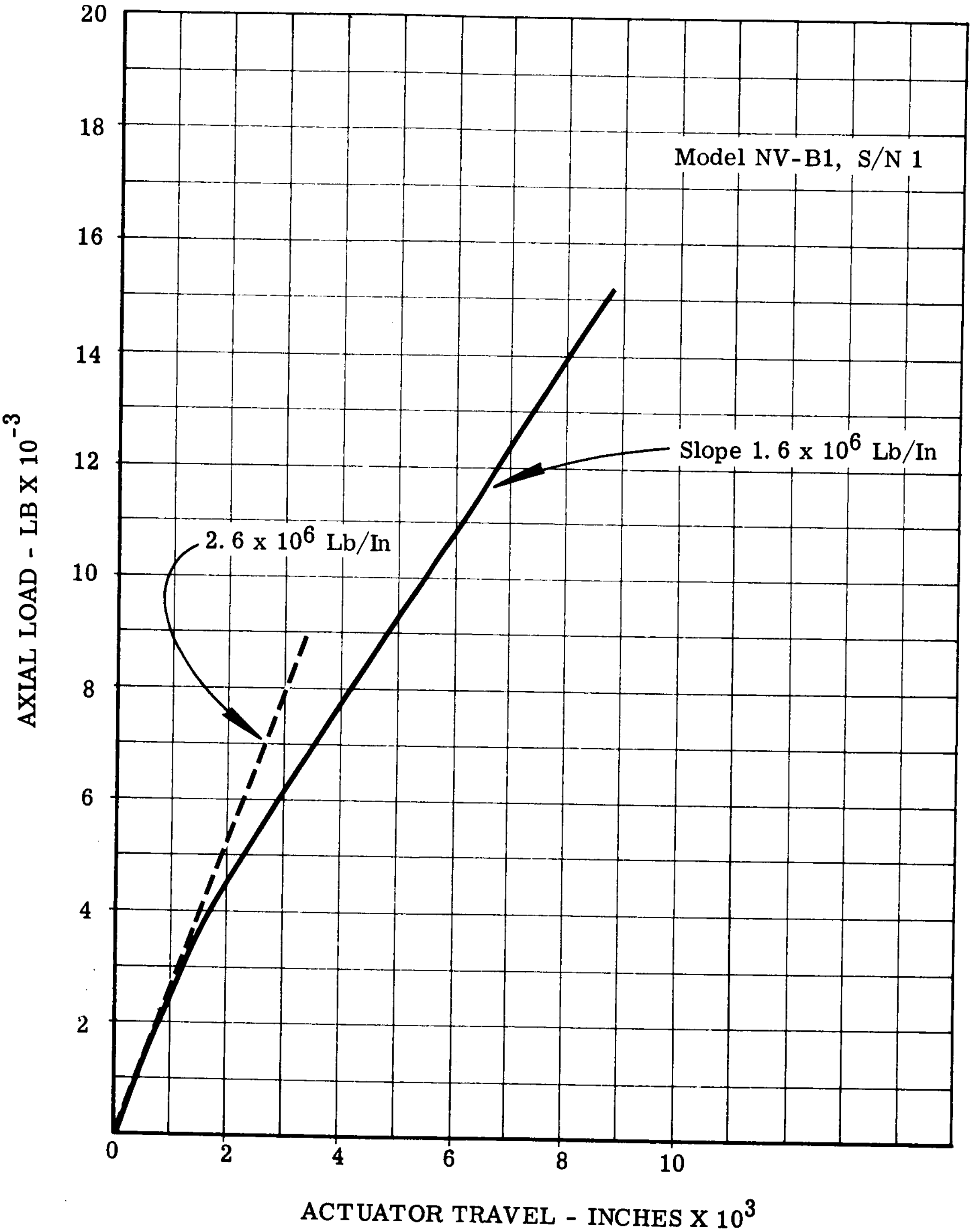


Figure 4-97. Actuator Mechanical Spring Rate.

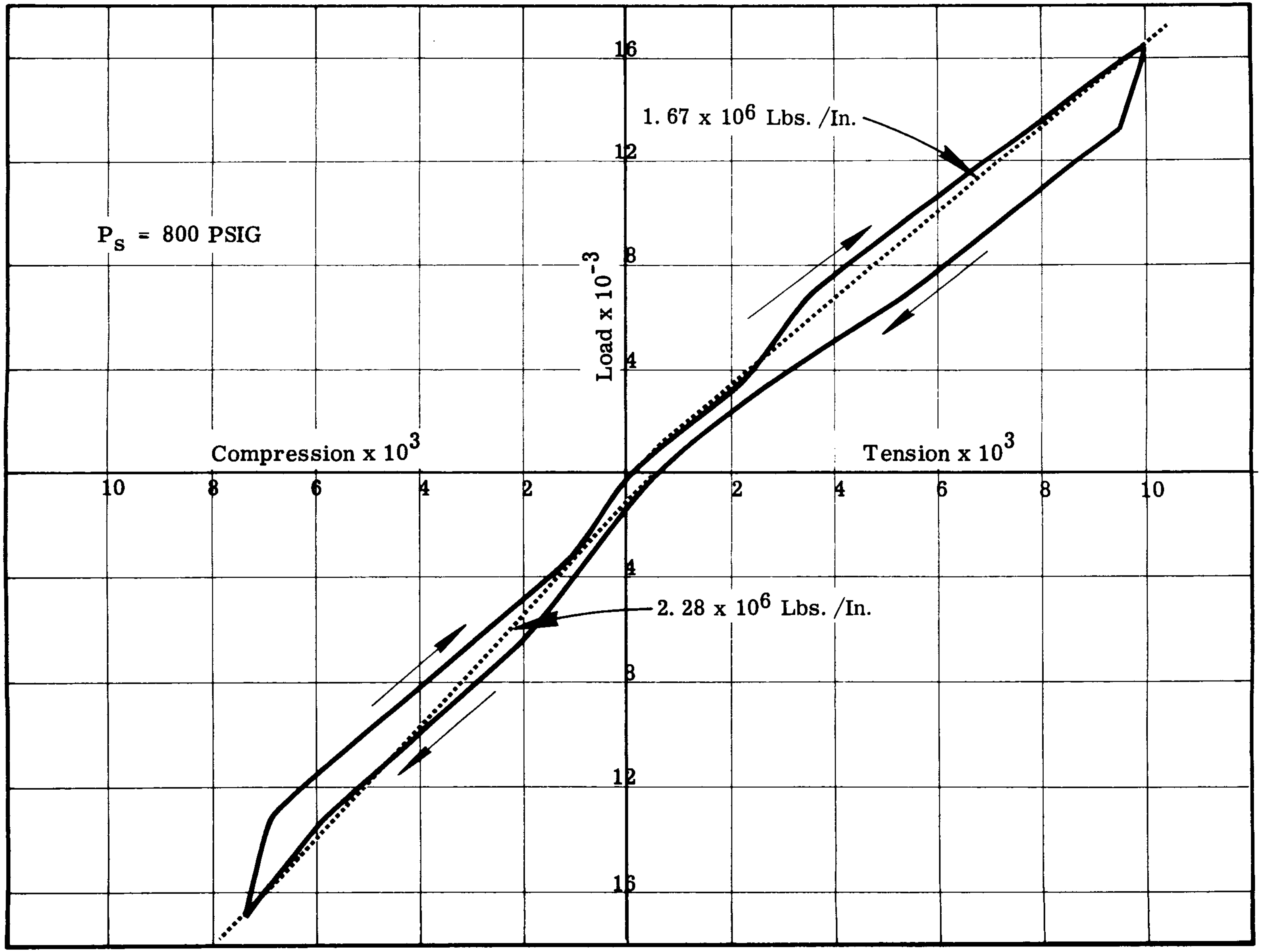


Figure 4-98. Closed Loop Static Stiffness.

actuator in the fixture would permit the vibrator axis to be through the plane perpendicular to the first plane and perpendicular to the ball screw centerline. To obtain the vibration axis parallel to the centerline of the ball screw, the fixture was placed on a film of oil on a granite slip table. The vibrator was rotated so that its axis would be parallel to the slip table surface and the fixture was attached directly to the vibrator armature.

Three accelerometers were attached directly to the actuator transmission bearing cover so that acceleration could be measured in all three planes. A fourth accelerometer was attached to the fixture at the point of input force with its principle axis on the axis of vibration. All four accelerometers and the vibration frequency were monitored by a five-channel recording oscillograph.

A thyratron circuit was connected to the actuator to monitor the feedback and position potentiometer. This setup indicates an open circuit with a light-on signal that must be reset once a circuit is broken.

The spool of the servo valve was blocked in a hard-over position since pressure was not available. This, in turn, positioned the torque motor in a hard-over position.

The complete vibration spectrum was scanned in all three planes and a continuous recording was made of all accelerometer and frequency data.

4.7.2 Summary of Results

Throughout the complete vibration runs there were no open circuits shown on the feedback potentiometer. A check on the torque motor after the vibration test indicated no change in its operational characteristics.

The only difficulty encountered during the testing was when vibrating through the axis perpendicular to a plane through the centerlines of the ball screw and pneumatic servo motor. The motor and transmission are displaced from the centerline and provide a substantial off-center moment. The test was started with the fixture stops loaded against the stabilizing lugs on the head end bracket eye, rather than providing the 0.010 to 0.023 inch clearance of the installation. During the constant 12G portion of the vibration spectrum, the fixture stops were failed by the moment created by the off-center motor and transmission. It was necessary to provide a spacer between the fixture and the head end bracket eye and draw down the stud through the spherical ball to provide sufficient contact area for this moment. (NOTE: This moment is approximately 20 lbs. x 5.875 inches - 117.5 in.-lbs. At 30G's this would be a moment of 3,520 in-lbs. or 2,450 lbs. force at the center of the stabilizing lug.)

The oscillograph traces indicated resonant conditions on one or more accelerometers as shown on Table 4-5, and Figures 4-99, 4-100, and 4-101.

Plane 1 with the axis of vibration perpendicular and through the centerline of the ball screw and servo motor indicated 15 resonant points. The most severe accelerations generated in comparison to driving acceleration were at 40 and 200 cps where 30G's were recorded along Axis 1 with a driving acceleration of 12G's.

Plane 1					
Trace No.	Freq. CPS	Input G's	1 G's	2 G's	3 G's
1	30	16	15	6	8
2	40	20	30	22	25
3	50	15	14	8	14
4	100	15	15	8	12
5	200	15	30	18	24
6	300	15	6	4	6
7	400	15	8	4	12
8	430	18	10	8	15
9	500	25	25	12	20
10	600	35	20	15	20
11	700	40	22	20	25
12	720	25	20	30	28
13	800	20	20	12	20
14	900	22	22	25	25
15	1000	25	8	6	12

Plane 3					
Trace No.	Freq. CPS	Input G's	1 G's	2 G's	3 G's
1	25	12	Accelerometer Dead	6	15
2	50	20		20	25
3	140	15		18	25
4	200	10		10	16
	275				
5	325	24		25	20
6	400	15		8	12
	500				
7	675	65		48	35
8	700	28		12	18
9	725	35		18	25
10	800	20		18	20
11	900	30		15	20
12	1000	22		10	16

Plane 2					
Trace No.	Freq. CPS	Input G's	1 G's	2 G's	3 G's
1	30	15	4	25	10
2	40	35	8	44	20
3	50	40	8	53	20
4	90	14	8	25	18
5	100	15	8	15	15
6	150	14	5	8	6
7	200	12	4	8	8
8	250	14	6	8	6
9	300	14	6	8	6
10	350	14	6	8	6
11	400	14	6	8	6
12	450	14	8	8	10
13	500	14	8	5	8
14	550	35	18	22	20
15	600	35	8	8	8
16	650	40	10	12	12
17	700	42	12	22	12
18	750	25	10	12	8
19	800	25	8	18	10
20	850	25	8	18	15
21	900	25	8	16	12
22	950	25	8	16	12
23	1000	25	8	16	12

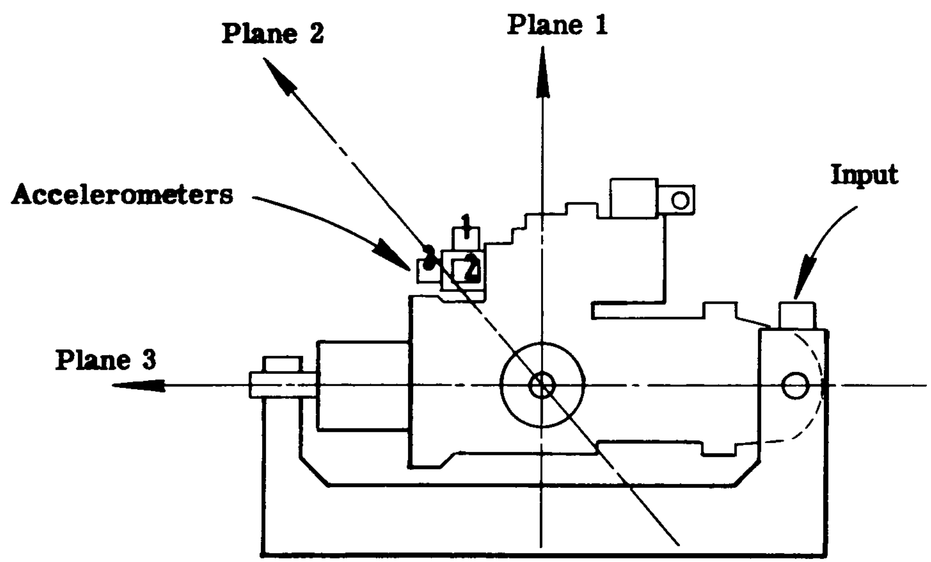


Table 4-5. Vibration Test Data - Actuator S/N-2.

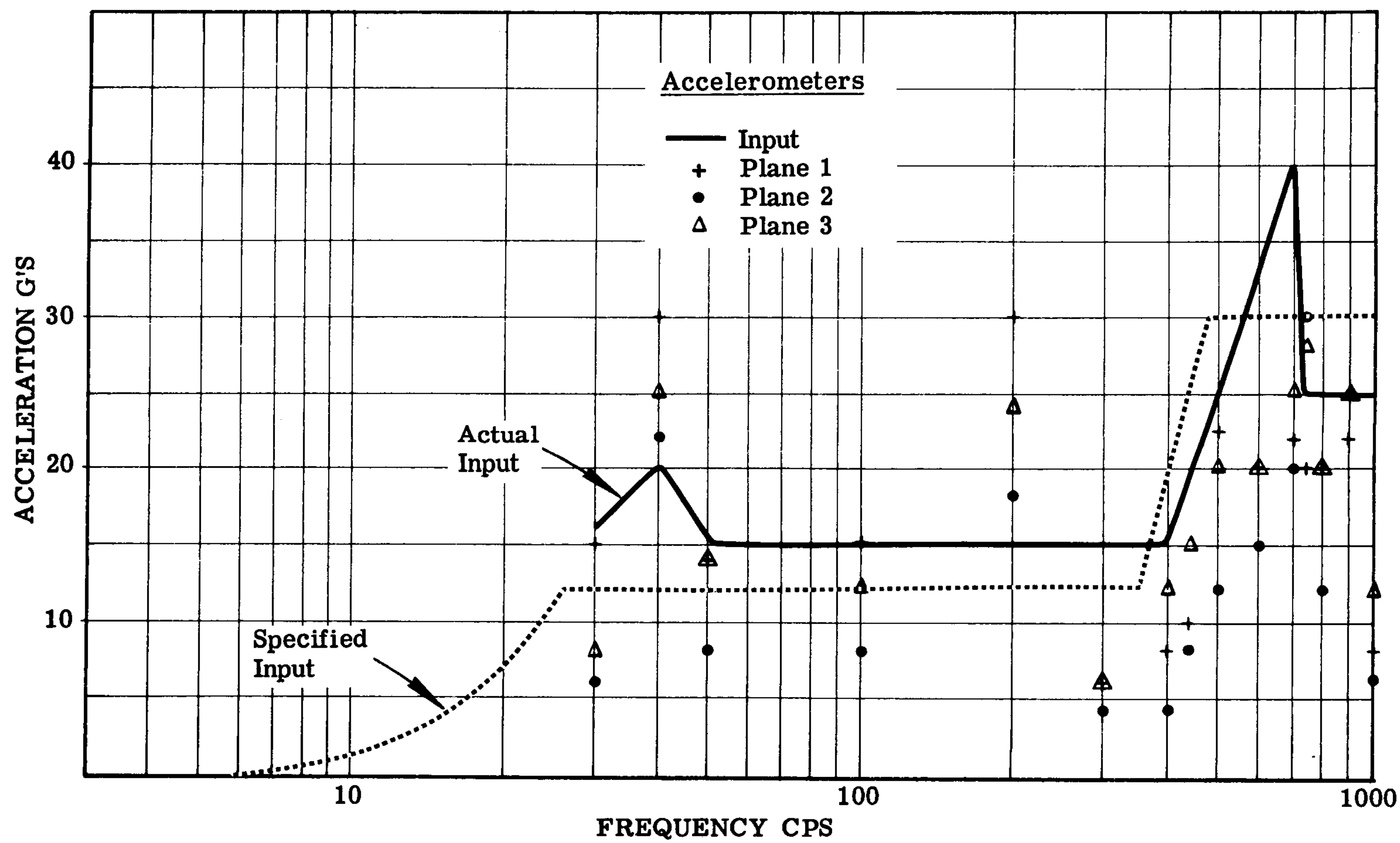


Figure 4-99. Plane 1 Accelerations.

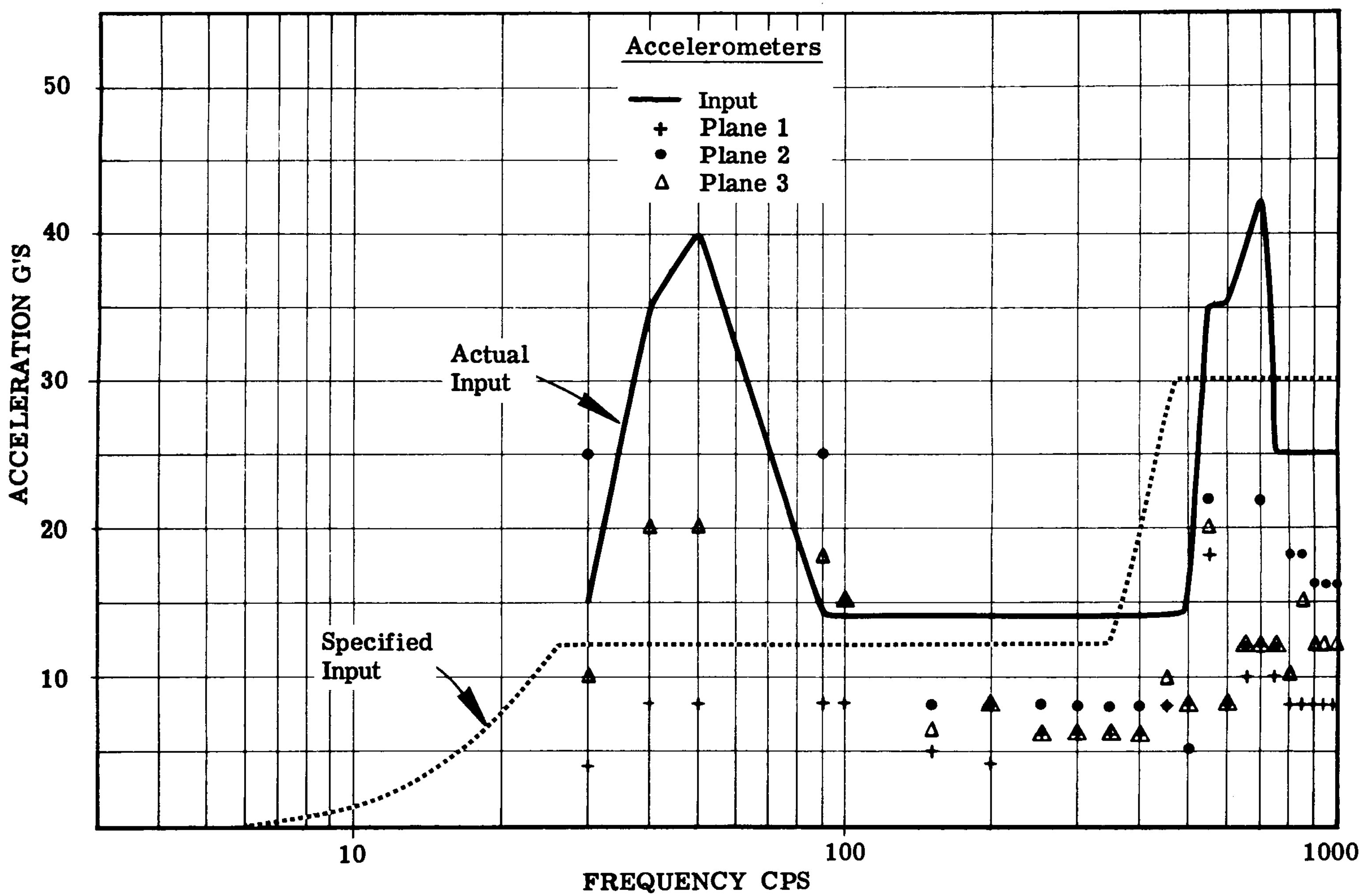


Figure 4-100. Plane 2 Accelerations.

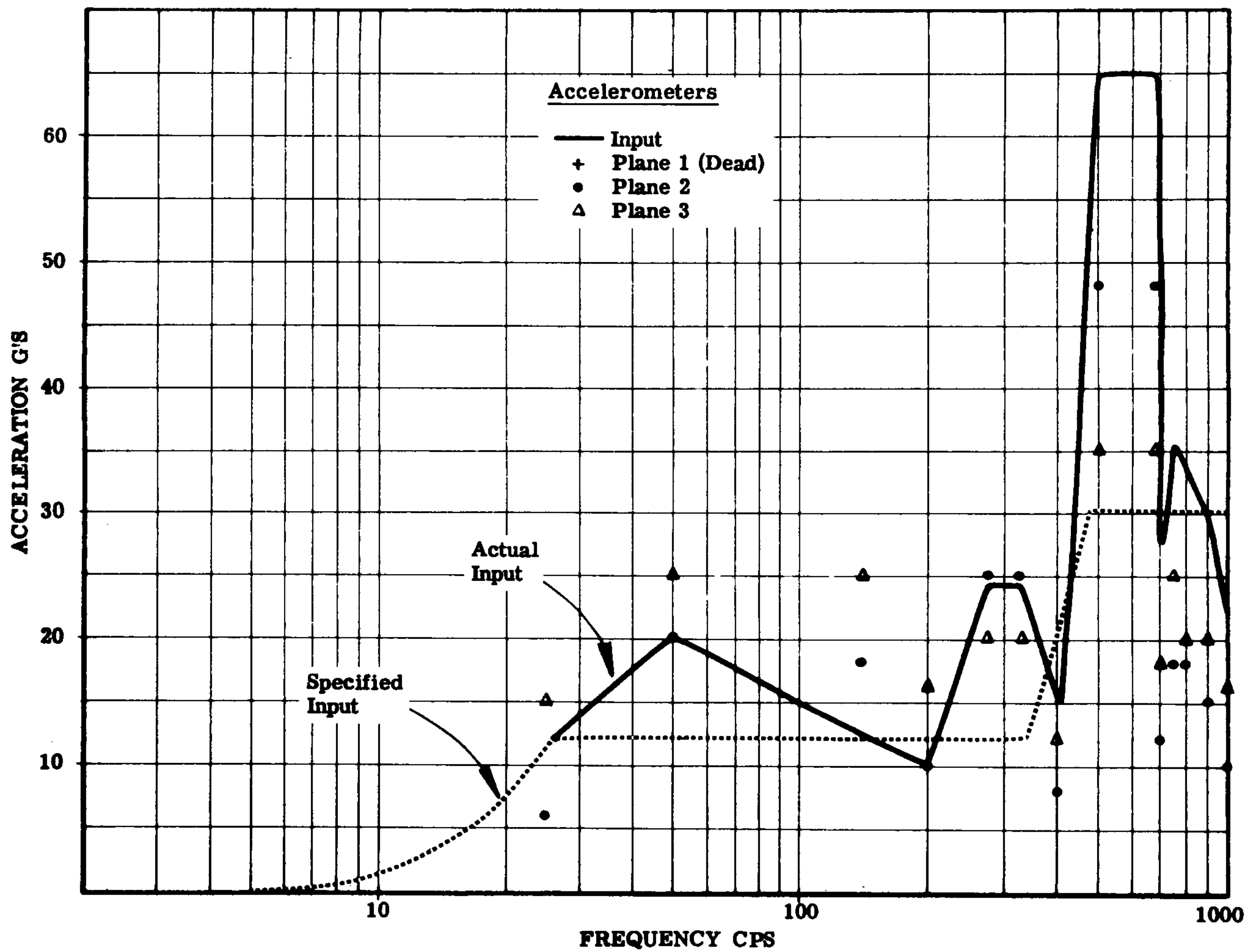


Figure 4-101. Plane 3 Accelerations.

Plane 2 with the axis of vibration perpendicular to a plane through the centerlines of the ball screw and servo motor indicated 23 resonant points. The most severe accelerations were generated in Plane 2 at 40 and 50cps with 44 and 53G's respectively. Except for the input accelerometer on the fixture, all resonant points were attenuated below driving accelerations at frequencies above 100cps.

Plane 3 with the axis of vibration parallel to the ball screw centerlines only 12 significant resonant points were recorded. There were only three points of high accelerations in the 12G range of the spectrum at 50 and 140 and between 275 and 325cps. The most severe of all resonances was found between 500 and 675cps where the fixture indicated accelerations as high as 65G's. Plane 2 was 48G's and Plane 3 on the actuator indicated 35G's.

4.7.3 Conclusions

It is concluded that the actuator was not damaged by the vibration tests. The changes made in the mounting of the potentiometer and its connector prevented discontinuities that were evident on preliminary vibration tests on the potentiometer (see section 4.2.6 of this report).

It is evident that the means of restraining the off-center mass by the lugs on the head end bracket eye will not be satisfactory for a vibration axis through Plane 2. This off-center loading was a possible cause for the high resonant accelerations generated in Plane 2 at 40 and 50cps.

The high resonant accelerations set up in Plane 3 between 500 and 675cps may have been caused by the means of securing the mass of actuator and fixture to the vibrator. It will be noted that Plane 2 indicated higher accelerations than Plane 3 and may have been a result of reaction torques produced by the lead screw against the loose stabilization stops specified because of spring rates of the components of the actuator.

4.7.4 Recommendations

It is recommended that consideration be given to a better method of stabilization stops capable of resisting the 3,520 in-lbs. torque developed by the off-center motor and transmission. A spherical ball ended strut connected between the transmission and the missile structure would be one possible solution. Another, better, solution would be a lengthened space envelope to permit the design of a symmetrical actuator if this would be compatible with system considerations.

Further vibration study should be conducted to determine the components that initiate the major resonant conditions found.

SECTION V

CONCLUSIONS

This report indicates that the general design approach taken on the Model NV-B1 Electropneumatic Actuator will provide a feasible thrust vector control for the J-2 engine.

The component tests run during this program indicate that the actuator was basically over-designed with respect to ball screw sizing. This was primarily because of the teflon ball method of lubrication developed after the ball screw had been sized. It should be pointed out that with the use of a smaller ball screw and use of aluminum or titanium for many of the structural parts that the weight of a follow-on design could be reduced to 50 to 60% of the NV-B1 design.

A basic error was made in the structural design. The large spanner nuts used on each end of the housing were intended to provide structural stiffness with a minimum number of parts. It was found, however, that friction on the large diameters required so much torque that the actuator had to be externally loaded to deflect the metallic "O" ring seals and the spanner nuts had to be torqued when in this condition.

Component testing of the dynamic reciprocating seal indicated that it would not provide a satisfactory seal at less than -170°F. For cryogenic operation, the inside lip of the seal should be used to seal against relative motion. Further development is required to obtain a satisfactory seal.

Component testing of the servo motor with the servo valve indicated a very high deadband to reverse the servo motor. The high deadband was a result of vane friction and cross-port leakage. The greatest percent of all development time was spent in an effort to reduce this deadband. Although a 40% improvement in deadband was attained during the development program, further work must still be done in this area. The servo motor requires additional development in order for the actuator to attain expected performance. The specification requirement of 0.25 ma cannot be obtained on a servo valve motor combination. It is believed that a total deadband of 3 to 4 ma with the gains involved would provide the expected actuator performance.

Most of the deficiencies in the closed loop performance of the control system can be attributed to the poor friction characteristics of the rotary vane motor. The overall resolution of the system was in the order of 0.010 to 0.012 inch which is not acceptable. The overall resolution could be improved by increasing the gain available from the servo amplifier, but this would only be obscuring the basic problem of poor vane motor resolution.

The closed loop frequency response of the system indicates that the dynamics of the actuator may be marginal if the 8 cps bandwidth is to be attained, particularly if the load friction force remains at 7,300 pounds. An improvement in the resolution of the motor would reduce the overall dynamic problem considerably.

The transient response was well within reason with respect to the rise time, overshoot, and settling time. A large portion of the overshoot was caused by the large lag in the amplifier which was necessary in order to be able to maintain a static resolution of the order required for system evaluation and still be reasonably stable. A reduction in this lag would improve the dynamic resolution as well as the transient response.

The snubber springs designed for the NV-B1 actuator were designed on the basis of maximum motor torque, full load and actuator inertia applied directly to the snubber springs with no energy absorption allowance for gear train and other components. The snubber spring deflection was therefore only 38 to 52 percent of the specification allowance. The snubbing forces could be reduced on a new design by reducing snubber spring rate to use a greater percentage of allowable snubber travel.

Vibration testing of the NV-B1 actuator design indicated that the method of mounting the actuator and supporting the off-center mass of motor and gear train with the stabilizing stops specified was not satisfactory. A spherical ball ended strut should be connected between the actuator and airframe to resist vibration forces of the off-center mass. A longer symmetrical envelope for the actuator would eliminate the need for the strut.

SECTION VI

RECOMMENDATIONS

Since feasibility of the NV-B1 type electropneumatic actuator has been shown, it would be the Bendix recommendation that the program be extended to design and fabricate flight-weight thrust vector actuators for the J-2 engine. A three phase program would be recommended as follows:

Phase I - Development programs on the servo motor and dynamic seals to provide better actuator resolution and increase reliability of the motor and to provide a seal capable of operation throughout the complete temperature range.

Phase II - Endurance test present NV-B1 actuators to determine actual sizing factors for ball screws, bearings, gears, etc. that may be used in a more optimum design. Study alternate designs and layout a flight-weight actuator.

Phase III - On approval of design of Phase II detail, fabricate, test and deliver two flight type actuators.

This program indicated several design considerations for similar future actuators as follows:

1. An actuator conforming to the space envelope of this specification requires a better method of supporting the unsymmetrical load involved than the raised stops specified on the head end bracket eye. A spherical ball ended strut between the actuator housing and the missile would be recommended.

2. If at all possible, the space envelope should be increased in length permitting the design of a symmetrical unit with the motor and transmission centerline on the centerline of the ball screw. An estimated six to eight inches additional length would be required.

3. All large spanner nuts must be securely locked against any possible torque reactions.

4. Linear potentiometers must be supported securely on both ends and internal electrical connectors must provide full pin engagement.

5. Ball splines must definitely trap balls within grooves of male and female splines.

6. Positive pressure feedback should be incorporated in the actuator in addition to phase lead.

APPENDIX A

PARTS LISTS

The NV-B1 Actuator is covered by parts list 2775024. It and the parts lists of major components are included in this appendix as follows:

2775024	NV-B1 Actuator
2153707	Servo Valve
2775037	Pneumatic Vane Motor Assembly

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY F. Gustafson

PL 2775024
DATE

PARTS LIST CHANGE INDEX

SERIAL NO.	DATE OF CHANGE	SHEET NUMBERS CHANGED	NATURE OF REVISION
1		1,2,3,4,5,6,7	EXPERIMENTAL RELEASE

LATEST CHANGE SERIAL NO. TO CORRESPONDING SHEET NO.

SHEET NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
SERIAL NO.	1	1	1	1	1	1	1																			

NAME **PNEUMATIC ACTUATOR (THRUST VECTOR)**

ENGRG DEPT **Advanced Dynamic Controls**
MODEL **NV-D1**
SHEET NO. 1 NO. OF SHEETS 7

MFR'S CODE NO. 06848

PL 2775024

A-1

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY

PL 2775024
DATE -

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1	1	2775024		ONE	PNEUMATIC ACTUATOR (THRUST VECTOR)			
2					(Composed of)			
3								
4								
5	2	2775037		ONE	MOTOR ASSY - VANE			
6								
7		2153711		ONE	VALVE & TORQUE MOTOR ASSEMBLY			
8	2	2153707		ONE	VALVE - SERVO ASSEMBLY			
9		2153963			TORQUE MOTOR ASSEMBLY			
10	3	2775034		ONE	AMPLIFIER ASSEMBLY			
11								
12		2775680		ONE	HOUSING ASSY - MAIN			
13		2775612		1	Housing - Main			
14		951266		1	Pin			
15								
16								
17								
18		2775663		ONE	BALL NUT, GEAR & BRACKET ASSY.			
19		2775564		ONE	BALL NUT & SCREW ASSY.			
20		2775587		1	Screw - Ball			
21		2775586		1	Nut - Ball			
22		2775588		1	Spline, Ball			
23		2775589		330	Ball, Bearing			
24		2775591		3	Tube, Ball Return			
25		2775590			Strap, Tube Clamp			
26		2775619		1	Retainer, Ball			
27		2775609		1	Gear			
28		951264		2	Pin, Straight Headless			
29		2775665		ONE	BRACKET EYE ROTATING SHAFT ASSY.			
30		2775613		1	Bracket Eye Rotating Shaft			
31		2775577		1	Bearing, Plain, Self-aligning			
32		951257		2	Pin, Straight Headless			
33								

NOTES
1 For Installation Drawing See 2775693
2 See Separate Parts List
3. System wiring diagram (see 2770237)

NAME PNEUMATIC ACTUATOR (THRUST VECTOR)

ENGRG DEPT Advanced Dynamic Controls
REL DATE
MODEL NV-B1
SHEET NO. 2
NO. OF SHEETS 7
SERIAL NO. See Sheet 1

MFR'S CODE NO. 06848

PL 2775024

A-2

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY GUSTAFSON

PL 2775024
DATE

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1								
2								
3		333-S-179		6	Screw, Dr.Soc.Hd.Cap (.250-28x.69)			
4		2775649		2	Plate, Retaining Pin			
5								
6								
7		2775639		1	Ring - Threaded (5.25 x 16)			
8		2775568		1	Bearing - Ball - Set			
9		2775640		1	Nut - Spanner			
10		2775694		1	Pin - Straight Headless			
11								
12								
13		2775610		1	Ring - Threaded (3.750 x 16)			
14		2775614		1	Bushing - Shoulder			
15		2775578		4	Disc - Spring			
16		2775615		1	Sleeve - Threaded			
17								
18								
19		2775594		1	Bushing - Rod End			
20		951872		1	Ring - Retaining			
21								
22		2775638		1	Retainer - Potentiometer			
23		333-S-574		6	Screw - Dr.Soc.Hd. Cap			
24		2775637		1	Guide - Spring			
25		2775655		1	Spring			
26		951866		1	Ring - Retaining			
27		2775620		1	Retainer - Spring			
28								
29		2775565		ONE	POTENTIOMETER ASSY.			
30		2775437		1	Potentiometer			
31		2775604		1	Connector			
32		2775642		1	Washer - Tab			
33		2775641		1	Nut			

NOTES

NAME PNEUMATIC ACTUATOR (THRUST VECTOR)

ENGRG DEPT Advanced Dynamic Controls
REL DATE MODEL NV-B1
SHEET NO. 3 NO. OF SHEETS 7
SERIAL NO. See Sheet 1

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MFR'S CODE NO. 06848

PL 2775024

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY

PL 2775024
DATE —

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1				ONE	WIRING HARNESS			
2		2775664		ONE	PLUG ASSY - CONNECTOR			
3		2775566		1	Plug			
4		2775643		1	Nut			
5		951174		1	Pin			
6		2775828		1	Spring			
7								
8		2775660		1	Receptacle, Connector			
9		2775659		1	Seal, Metallic			
10		333-S 502		4	Screw, Dr. Soc. Hd. Cap (#4-40 x .25)			
11								
12								
13		2775677		1	Packing Metallic			
14		2775648		1	Key			
15		2775618		1	Nut - Spanner (6.00 x 16)			
16								
17								
18		2775585		ONE	BUSHING ASSY.			
19		2775584		1	Bushing			
20		336066		1	Insert, Heli-coil (mid grip)			
21		951174		1	Pin, Straight Headless			
22		2775647		1	Washer, Key			
23		2775658		1	Nut - Lock			
24		2775646		1	Nut-Spanner			
25		2775672		1	Ring - Retaining			
26		2775595		1	Bearing Bushing			
27		2775673		1	Packing Preformed			
28		2775657		1	Seal, Metallic			
29		2775593		1	Spacer			
30		2775675		1	Packing, Preformed			
31		2775567		1	Bearing, Plain, Rod End			
32		2775592		1	Nut			
33		2775681		As req'd	Plastic Bearing Material			

NOTES

NAME PNEUMATIC ACTUATOR (THRUST VECTOR)

ENGRG DEPT Advanced Dynamic Controls
REL DATE MODEL NV-B1
SHEET NO. 4 NO. OF SHEETS 7
SERIAL NO. See Sheet 1

MFR'S CODE NO. 06848

PL 2775024

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY

COMPILED BY

PL 2775024

DATE

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1		2775616		1	Cover			
2		2775676		1	Packing, Metallic			
3		2775645		As reqd.	Shim			
4		2775617		1	Nut, Spanner (7.375-16)			
5		951200		1	Pin, Straight Headless			
6								
7		2775652		1	Spacer			
8		2775575		1	Gear, Idler			
9		2775580		2	Bearing			
10		2775576		1	Spacer			
11		2775651		1	Shaft, Pin			
12								
13								
14		2775582		1	Bearing			
15		951817		1	Ring, Retaining			
16								
17								
18								
19		2775668		1	Bracket, Indicator			
20		2775667		1	Indicator, Scale			
21		2775666		1	Scale, Vernier			
22		333-S-571		8	Screw, Dr. Soc. Hd. Cap (#6-32x.31)			
23		333-S-618		2	Screw, Dr. Soc. Hd. Cap (#8-32x1.25)			
24		901712-K1		2	Nut, Hex Thin. (.250-28)			
25		2775438		1	Clamp, Rim Clenching			
26		2775439		1	Clamp, Band			
27		151784		1	Plug & Bleeder			
28		2775669		1	Name Plate			
29		2775679		1	Packing Preformed			
30		2775678		1	Packing Metallic			
31		2775674		1	Seal Metallic			
32		2775671		1	Fitting 90° Elbow			
33		333-S-142		4	Screw, Dr. Soc. Hd. Cap (#10-32 x .50)			

NOTES

NAME PNEUMATIC ACTUATOR (THRUST VECTOR)

ENGRG DEPT Advanced Dynamic Controls
REL DATE
MODEL NV-B1
SHEET NO. 5
NO. OF SHEETS 7
SERIAL NO. See Sheet 1

A-5

MFR'S CODE NO. 06848

PL 2775024

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY Gustafson

PL 2775024
DATE

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1		2775569		ONE	TRANSMISSION ASSY.			
2		2775573		ONE	GEAR ASSY. - INTERNAL RING			
3		2775600		1	Gear - Ring			
4		2775601		1	Flange - Gear			
5		951227		2	Pin - Dowel			
6		2775602		2	Retainer - Dowel			
7		333-S-148		6	Screw-Dr.Soc.Hd.Cap (#10-32x.875)			
8								
9								
10								
11		2775570		ONE	GEAR ASSY - OUTPUT			
12		2775608		1	Carrier - Planet Gear			
13		2775607		1	Gear - Output			
14		951225		3	Pin-Dowel			
15		2775606		1	Retainer - Dowel			
16		333-S-144		3	Screw-Dr.Soc.Hd. Cap (#10-32 x .625)			
17								
18								
19		2775571		3	Gear - Planet			
20		2775581		6	Bearing - Ball			
21		2775576		6	Spacer			
22		2775574		3	Shaft - Bearing			
23		116559		3	Pin - Cotter			
24								
25								
26		2775572		1	Gear - Sun			
27								
28		951854		1	Ring - Retaining			
29		2775656		1	Seal - Metallic			
30		2775653		1	Cover			
31		333-S-143		6	Screw-Dr.Soc.Hd. Cap (#10-32x.5625)			
32								
33								

NOTES

NAME PNEUMATIC ACTUATOR (THRUST VECTOR)

ENGRG DEPT Advanced Dynamic Controls
REL DATE
MODEL NV-B1
SHEET NO. 6
NO. OF SHEETS 7
SERIAL NO. See Sheet 1

MFR'S CODE NO. 06848

PL 2775024

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY ... Miller

PL 2775024
DATE

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1		MS-20995C20 (629814)		As reqd.	Lockwire			
2		MS-20995C62 (163064)		As reqd.	Lockwire			
3		MS-20995C32 (159689)		As reqd.	Lockwire			
4								
5								
6					THE FOLLOWING PARTS ARE USED FOR			
7					SHIPPING & STORAGE PURPOSES			
8								
9		AN929-8C (911522)			Cap - Pressure Seal			
10		AN814-10D (158473)			Plug & Bleeder - Screw Thread			
11		557-8-10			Packing, Preformed			
12		175681			Cap Protective			
13								
14								
15								
16								
17								
18								
19								
20								
21								
22								
23								
24								
25								
26								
27								
28								
29								
30								
31								
32								
33								

NOTES

NAME PNEUMATIC ACTUATOR (THRUST VECTOR)

A-7

ENGRG DEPT Advanced Dynamic Controls
REL DATE MODEL NV-B1
SHEET NO. 7 NO. OF SHEETS 7
SERIAL NO. See Sheet 1

MFR'S CODE NO. 06848

PL 2775024

PART NO. 8808-000-179

CONTRACT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN **PARTS LIST**	SHEET 1 OF —	PARTS LIST NO.	REV.
		CODE IDENT. NO. **11272**	2153707	
CUSTOMER AND CUSTOMER NO.: Bendix Products Aerospace Div		DATE ISSUED: 12 Dec 63	PROJECT NO.: 3312-301	

NAME: Assembly - J2 Servo Valve, model No. A

NEXT ASSY NO.

ITEM NO.	DWG. SIZE	BENDIX PART NO.	REV. LET.	OTHER PART NO.	QTY. REQ'D	PART NAME	CIRCUIT SYMBOL
1	D	2153707			1	Assembly - J2 Servo Valve, model [illegible]	
2							
3							
4	E	2153531 △3			1	Valve Body - J2 Servo Valve	
5	E	2153531-1			1	Body	
6	E	2153531-2			1	Plug	
7							
8							
9							
10	C	2153611 △3 △1			1	Spool & Nozzle Assy - J2 Servo Valve	
11	C	2153496 △4			1	Spool Assy - Brazed	
12	C	2153496-1			1	Spool	
13	C	2153496-2			2	Plug	
14							
15							
16	C	2153495 △4			1	Nozzle Assy - J2 Servo Valve	
17	C	2153495-1			1	[illegible]	
18	C	2153495-2			2	Nozzle	
19							
20							
21							
22	B	2153565			1	Sleeve - Spool Bearing, J2 Servo [illegible]	
23							
24							
25							
26	C	2154077			2	Damping Tank	
27	C	2153532			2	Spacer Plate Assy	
28	C	2153532-1			1	Spacer Plate [illegible]	
29	C	2153532-2			1	dowel pin	
30							

NOTES

△1 Ref Spec BRLD Spec No. [illegible]

△2 Ref [illegible]

△3 Ref [illegible] BRLD [illegible]

△4 Ref Spec BRLD [illegible]

ORIGINATED BY	DATE	CHECKED BY	DATE
APPROVED	DATE	APPROVED	DATE

BC RLD-238

PART NO. 8808-000-179

CONTRACT NO.	CUSTOMER AND CUSTOMER NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	SHEET 2 OF 2 CODE IDENT. NO. 11272	PARTS LIST NO. 2153707	REV.
		PARTS LIST	DATE ISSUED	PROJECT NO.	
NAME				NEXT ASSY NO.	

ITEM NO.	DWG. SIZE	BENDIX PART NO.	REV. LET.	OTHER PART NO.	QTY. REQ'D	PART NAME	CIRCUIT SYMBOL
1	C	2153532-3			1	Spacer Plate - R.H.	
2	C	2153532-2			1	Dowel Pin	
3							
4							
5		STK			2	Seal-metallic Hydrodyne No 1100-	
6							
7							
8		STK			10	Scr-#10-32 UNF-3A x .56 lg Soc Hd Cap	
9							
10							
11							
12	C	2153566			1	Plate Assy - Seal Retaining, 32 Servo Valve	
13	C	2153566-1			1	Plate	
14	C	2153566-2			2	Dowel Pin	
15							
16							
17							
18							
19							
20	C	2153640		Ref Print		FLOW TRIM ASSY	
21							
22							
23							
24							
25							
26							
27							
28							
29							
30							

NOTES	ORIGINATED BY	DATE	CHECKED BY	DATE
	APPROVED	DATE	APPROVED	DATE

BC/RLD-238

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

PARTS LIST CHANGE INDEX

CHECKED BY
COMPILED BY L. Miller

PL 2775037
DATE 6-16-64

SERIAL NO.	DATE OF CHANGE	SHEET NUMBERS CHANGED	NATURE OF REVISION
1	6-16-64	1, 2, 3	Experimental Release

LATEST CHANGE SERIAL NO. TO CORRESPONDING SHEET NO.

SHEET NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
SERIAL NO.	1																									

NAME PNEUMATIC VANE MOTOR ASSEMBLY

ENGRG DEPT Advanced Dynamic Controls
MODEL
SHEET NO. 1 NO. OF SHEETS 3

MFR'S CODE NO. 06848 PL 2775037

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY **L. Miller**

PL **2775037**
DATE **6-12-64**

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1		2775037		ONE	PNEUMATIC VANE MOTOR			
2					ASSEMBLY			
3								
4								
5								
6		2153980		ONE	BODY - BRAZED ASSY.			
7		2153980-1		1	Body			
8		2153980-2		1	Pin			
9		2153985		1	Insert - Porting			
10				2	Pin, Straight Headless			
11		MS 21209-F415		28	Insert			
12		MS 21209-F115		4	Insert			
13								
14								
15								
16		2775684		ONE	END CAP, ASSEMBLY REAR			
17		2775688		1	End Cap, Rear			
18		2775682		1	Bushing			
19		2775685		1	Washer End Cap Facing			
20								
21								
22		2775683		ONE	END CAP, ASSEMBLY PILOT			
23		2775687		1	End Cap, Pilot			
24		2775682		1	Bushing			
25		2775686		1	Washer End Cap Facing			
26								
27								
28		2154004		1	Retainer - Bearing			
29		2154001		1	Plate - Spacer			
30		2154005-1		1	Retainer, Dowel			
31								
32								
33								

NOTES

NAME **Pneumatic Vane Motor Assembly**

ENGRG DEPT **Advanced Dynamic Controls**
REL DATE MODEL
SHEET NO. **2** NO. OF SHEETS **3**
SERIAL NO.

MFR'S CODE NO. 06848

PL 2775037

A-11

THE Bendix CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND 20, INDIANA, U.S.A.

PARTS LIST

CHECKED BY
COMPILED BY L. Miller

PL 2775037
DATE 6-12-64

ITEM	NOTE	PART NUMBER	CUSTOMER PART NO.	NO. REQD	NAME	CHANGE SERIAL NO.	NO. REQD LH	NO. REQD RH
1		2153984		1	Rotor			
2		2153996		9	Spring, Flat Leaf			
3		2775689		9	Vane			
4		2153999		1	Retainer, Bearing			
5		2154005-2		1	Retainer, Dowel			
6		2153997		1	Sleeve, SunGear			
7		2153998		1	Washer, Retaining			
8		2153995		1	Nut			
9		2154055		1	Nut			
10		2154003		4	Spacer			
11								
12								
13		2775583		2	Bearing, Ball			
14								
15		379-S-145		1	Seal, Metallic			
16		2775691		1	Seal, Metallic			
17		2775692		1	Seal, Preformed			
18								
19								
20		952654		4	Screw, Soc. Hd. Cap (#4-40 x .31)			
21		952963		8	Screw, Soc. Hd. Cap (#10-32 x .50)			
22		953029		12	Screw, Soc. Hd. Cap (.250-28 x .62)			
23		953033		12	Screw, Soc. Hd. Cap (.250-28 x .88)			
24		2154002		4	Bolt, 12 Point			
25								
26								
27		2775288		1	Filter			
28								
29		2775690		1	Washer, Tab Lock			
30								
31		2775572		1	Gear, Planetary Drive			
32								
33								

NOTES

NAME PNEUMATIC VANE MOTOR ASSEMBLY

ENGRG DEPT Advanced Dynamic Controls
REL DATE
MODEL
SHEET NO 3 NO. OF SHEETS 3
SERIAL NO.

MFR'S CODE NO. 06848 PL 2775037

APPENDIX B

SPECIFICATIONS

The NV-B1 Actuator was designed to meet the requirements of George C. Marshall Space Flight Center Specification 50M35003. In order to obtain components for the actuator to meet this specification, the following specifications were written and are included in this appendix.

NPD-110	Feedback Potentiometer (BPAD)
NPD-111	Servo Amplifier (BPAD)
DS-659	Torque Motor for Gimbal Valve Application (RLD)
DS-668	Servo Valve No. 3812 (J-2 Actuator) (RLD)
DS-676	Pneumatic Rotary Motor (RLD)

Bendix

PROJECT NO.	THE BENDIX CORPORATION BENDIX PRODUCTS AEROSPACE DIVISION SOUTH BEND, INDIANA	SPECIFICATION NO. NPD-110	A

ENGINEERING SPECIFICATION

TITLE	DATE
FEEDBACK POTENTIOMETER	May 17, 1963

1.0 POTENTIOMETER TYPE

The potentiometer shall be of the infinite resolution type, having dual elements and center taps.

2.0 ENVIRONMENT

The potentiometer shall be capable of meeting all performance requirements under the following environmental conditions:

Temperature....................-250° F to +150° F
Pressure......................10^{-6} mm Hg to sea level
Vibration.....................40 g's, 100 to 1000 cps
Atmosphere....................Gaseous Hydrogen

3.0 POTENTIOMETER LIFE

The potentiometer shall be capable of meeting all performance requirements under the environment conditions of Item 2.0 at any time during 130,000 operating cycles at full stroke and 0.5 cps. The ambient temperature and pressure during the life cycle shall be 77° F and one (1) atmosphere respectively.

The potentiometer shall be capable of meeting all performance requirements under the environmental conditions of Item 2.0 after a 30-day soak at -250° F and 10^{-6} mm Hg, and after an 8-hour soak at +150° F and one (1) atmosphere.

4.0 POTENTIOMETER SUPPLY VOLTAGE

The voltage across each potentiometer element shall be 60 vdc continuous. The potentiometer shall be capable of withstanding 90 vdc to both elements for a period of one (1) hour at an ambient temperature of 77° F without degrading its performance at the specified environmental conditions.

5.0 POTENTIOMETER POWER DISSIPATION

The total power supplied to both elements shall not exceed 3.75 watts under the environmental conditions of Item 2.0 and with a supply voltage of 60 vdc.

PREPARED BY	CHECKED BY	APPROVED BY
B. R. Coleman BRC	B. Hegg BH	D. J. Schaffer

REVISIONS (A) Exp. Released 7-1-63

ORIGINAL FILED IN PRODUCT DESIGN SECTION

6.0 ELEMENT RESISTANCE

The resistance of each element shall be a minimum as defined by Items 4.0 and 5.0 but shall not exceed 2500 ohms.

7.0 LINEARITY

The voltage ratio output (E_o/E_s) shall be linear with displacement to within the allowable error band shown in Figure II. E_o and E_s are defined in Figure I.

8.0 ELECTRICAL STROKE

The electrical stroke shall be 3.15 ± 0.020 inches minimum.

9.0 MECHANICAL STROKE

The mechanical stroke shall be 3.5 ± 0.020 inches.

10.0 WIPER DISCONTINUITY

There shall be no discontinuity in the output voltage (V_o) over the center 95 percent of the mechanical travel when the potentiometer is loaded with a 25,000 ohm resistor between the center tap and the element wiper output terminal and with 60 vdc as the supply voltage for the environmental conditions specified in Item 2.0.

11.0 PROTECTIVE RESISTORS

A 500 ± 50 ohm, 1/8 watt resistor shall be incorporated in series with each wiper and within the potentiometer housing. When resistance is checked through wiper, resistance shall be between 450 & 700 OHMS.

B. R. Coleman BRC cb	B. Hegg	D. J. Schaffer

(A) Exp. Released 7-1-63
(B) Resistance between 450 & 700 OHMS, Added 9-30-63

PROJECT NO.	THE BENDIX CORPORATION BENDIX PRODUCTS AEROSPACE DIVISION SOUTH BEND, INDIANA	SPECIFICATION NO. NPD-110	A

ENGINEERING SPECIFICATION

TITLE FEEDBACK POTENTIOMETER	DATE May 17, 1963

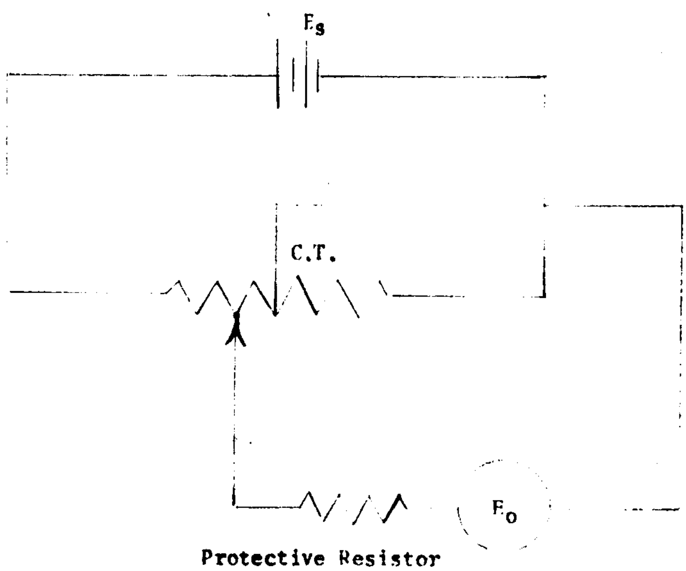

FIGURE I - POTENTIOMETER TEST CIRCUIT

PREPARED BY	CHECKED BY	APPROVED BY
B. R. Coleman BRC	B. Hegg BH.	D. J. Schaffer DJS

REVISIONS (A) Exp. Released 7-1-63

ORIGINAL FILED IN PRODUCT DESIGN SECTION

THE BENDIX CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND, INDIANA

NPD-110 A

ENGINEERING SPECIFICATION

FEEDBACK POTENTIOMETER

May 17, 1963

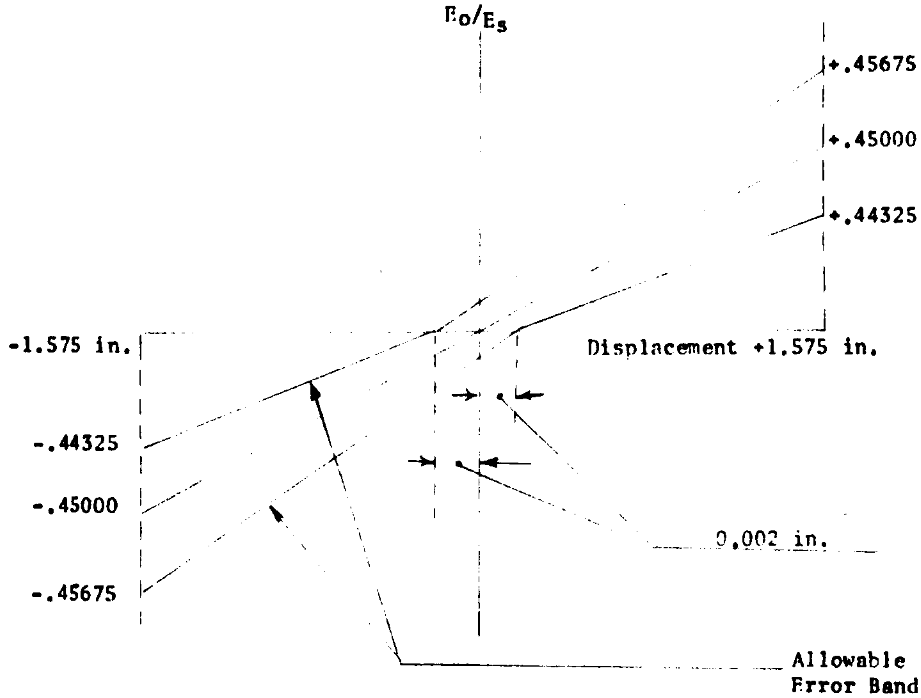

FIGURE II - POTENTIOMETER LINEARITY

PREPARED BY	CHECKED BY	APPROVED BY
B. R. Coleman BRC	B. Hegg BH.	D. J. Schaffer DJS

REVISIONS: (A) Exp. Released 7-1-63

ENGINEERING SPECIFICATION

TITLE: SERVO AMPLIFIER

DATE: September 13, 1963

1.0 Type

The amplifier shall be designed utilizing solid state components, preferably transistors. It shall be designed for a supply voltage of 115 volts ±5%, 60 or 400 cps ±5%, and for push-pull type of operation.

2.0 Inputs

The amplifier shall be designed to accept two independent grounded DC signals, and one floating differential DC signal

3.0 Signal Source Impedance

The external signal source impedance will be a maximum of 3000 ohms resistive.

4.0 Maximum Signal Level

The maximum signal level will not exceed ±60 volts DC.

5.0 Amplifier Input Impedance

The amplifier input impedance shall be greater than 100,000 ohms with or without supply power to the amplifier.

6.0 Amplifier Load

The amplifier load will consist of a two-wire, parallel winding, permanent magnet torque motor with a DC resistance of 100 ohms at 77° F and a maximum current rating of ±50 milliamperes. The torque motor will be subjected to an ambient temperature range of -250° F to +150° F. For amplifier evaluation, the load shall consist of a 2.4 henry inductor in series with a 116-ohm or a 28.5-ohm resistor.

7.0 Amplifier Dynamic Load Impedance

The load impedance and phase shift as a function of frequency and temperature is shown in Figure 1.

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RRC cb B. R. Coleman		D. J. Schaffer

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(B) Para 2.0 Read "The amplifier shall be designed to accept three independent insolated balanced external DC signals. 10-9-63

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8.0 Amplifier Static Gain

The amplifier maximum static gain through null shall be 0.070 ±10% amps/volt with static loads as defined in Figure 1.

9.0 Amplifier Static Gain Adjustment

The amplifier shall be provided with a readily accessible uncalibrated continuous static gain adjustment. The gain shall be adjustable over a 5:1 range. The frequency response or the output current saturation level shall not be affected by the gain setting.

10.0 Amplifier Frequency Response

The amplifier shall be designed to provide two basic frequency responses with a load as defined in Items 5.0 and 6.0, and with an input signal corresponding to ±25 milliamperes of output current. The input signal source impedance shall be as defined in Item 3.0. The frquency responses shall be as follows.

10.1 The ratio of output current to input voltage shall be flat from 0 to 150 cps minimum. The phase shift at 100 cps shall be no greater than -20°.

10.2 The amplifier shall be provided with readily accessible partially connected phase lead and phase lag circuits which when connected will result in the nominal frequency response as shown in Figure 2. The break frequencies of the phase lead and phase lag circuits shall be adjustable by means of component changes over a frequency range of ± one octave from the nominal. The amplifier shall have no internal instability when the break frequencies are adjusted over this range.

11.0 Linearity

The static gain curve shall be a smooth curve from minus saturation to plus saturation, and the slope of the curve shall not deviate more than ±5% of the gain through null from -75 to +75% of saturation. The signal required to drive the amplifier from 75% of saturation to 95% of saturation shall not exceed 35% of the total signal required for 95% of saturation.

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TITLE
SERVO AMPLIFIER

DATE
September 13, 1963

12.0 Offset

The output current with zero input voltage shall not exceed ±1.0 milliampere.

13.0 Ripple

The output current ripple shall not exceed 0.01 milliamperes peak-to-peak.

14.0 Saturation

The maximum output current shall be 50 +10 or -5 milliamperes.

15.0 Resolution

There shall be no hysteresis or deadband exhibited by the amplifier.

16.0 Drift

The amplifier drift shall not exceed 0.05 milliamperes of output current per hour at an ambient temperature of 77° F after a 30-minute warm-up period.

17.0 Delay Time

The amplifier shall exhibit no delay time from input signal to output current when the first stage of the amplifier is driven from saturation to saturation.

18.0 Environment

The amplifier shall conform within this specification at an ambient temperature of 77° F ±10° F, and an ambient pressure of one atmosphere.

19.0 Test Data

Test data to substantiate conformance with this specification shall be submitted for Bendix approval prior to acceptance of the amplifier.

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THE BENDIX CORPORATION
BENDIX PRODUCTS AEROSPACE DIVISION
SOUTH BEND, INDIANA

NPD-111 | A

ENGINEERING SPECIFICATION

SERVO AMPLIFIER

September 13, 1963

20.0 **Wiring Diagram**

A complete amplifier wiring diagram shall be supplied upon delivery of the amplifier.

21.0 **Connector**

The amplifier chassis shall be provided with a standard AN connector for all external connections.

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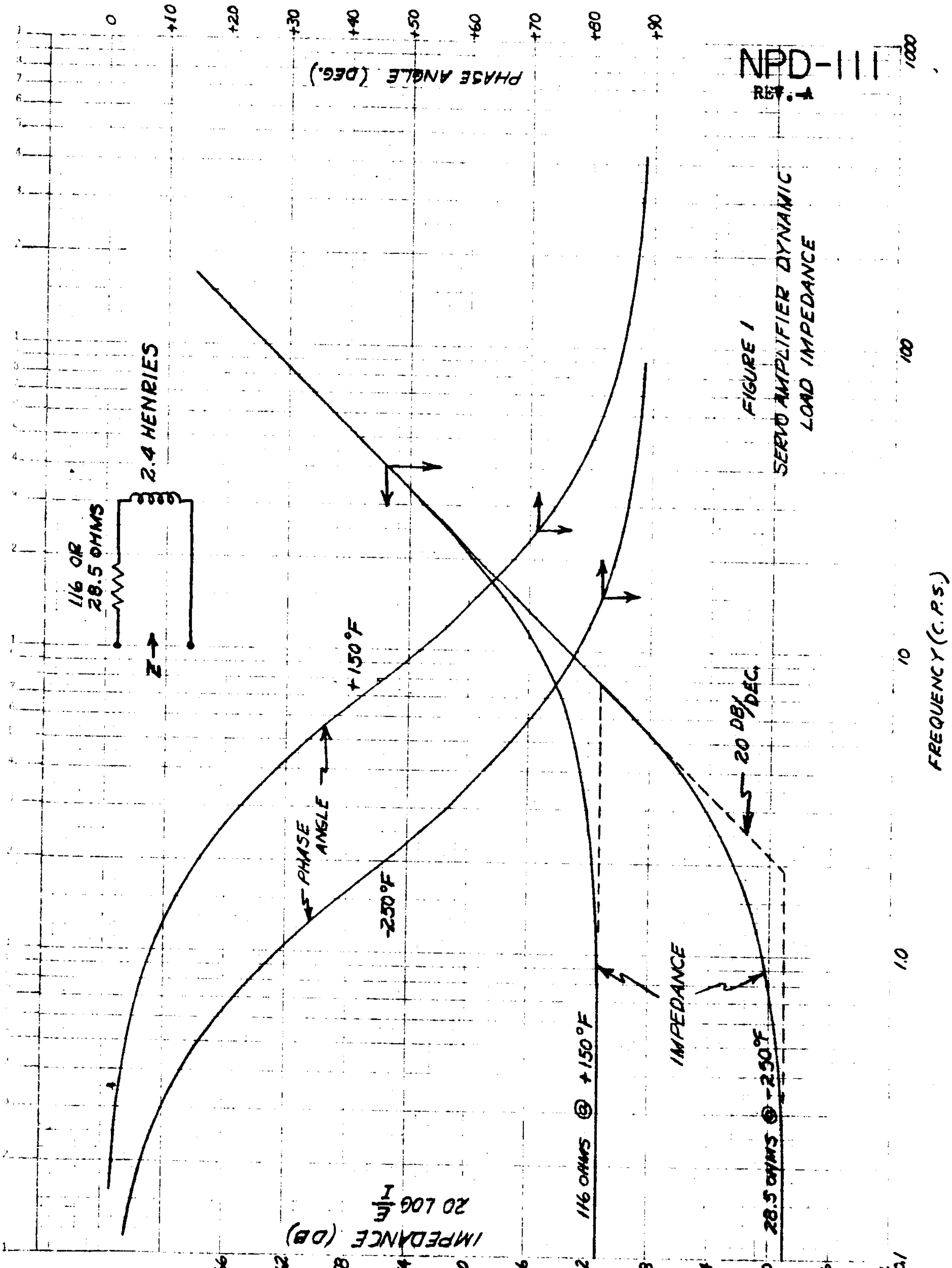

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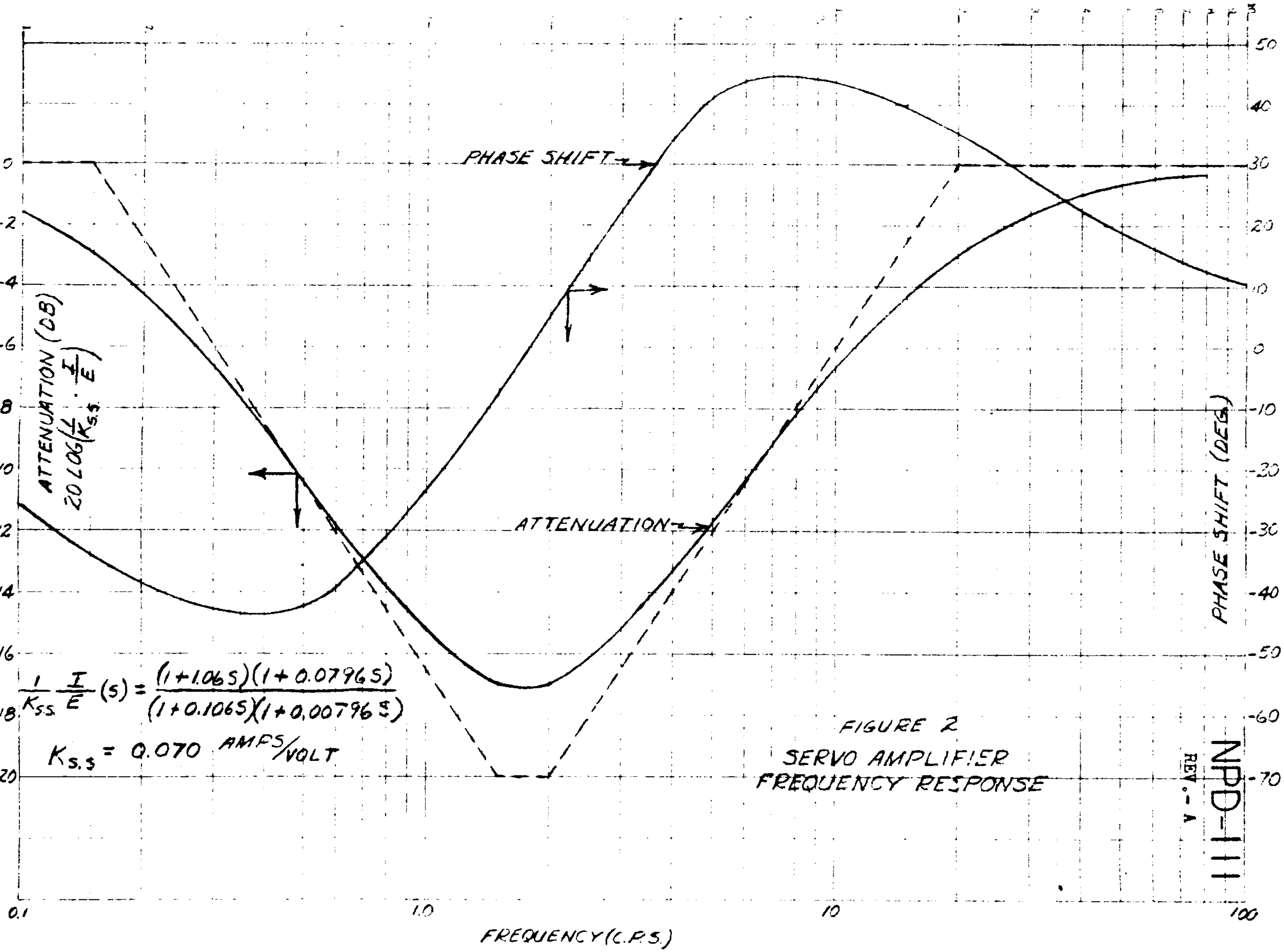

FIGURE 2
SERVO AMPLIFIER
FREQUENCY RESPONSE

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3812-100		11272	DS- 659	A

ENGINEERING SPECIFICATION

TITLE	DATE
Torque Motor for Gimbal Valve Application	8-14-63

Output Characteristics	Value	
Output point on shaft, below flat mounting base (see Figure 2).	1.000	inch
Output displacement at rated current, parallel to base	±0.015	inch
Maximum output displacement	±0.020	inch
Force at rated output displacement at max. current	3.0	ounces
Linearity within rated output displacement	±10.0	% max.
Displacement gain variation with temperature	±5.0	% max.
Hysteresis	2.0	% of rated current, max.
Resolution	0.5	% of rated current, max.
Null shift with temperature and pressure	1.0	% of rated current, max.
Frequency response to constant current source should approximate a first order lag with a 45° phase shift at	100.0	cycles/sec.
At 10 cycles/sec., maximum allowable phase shift	20.0	degrees

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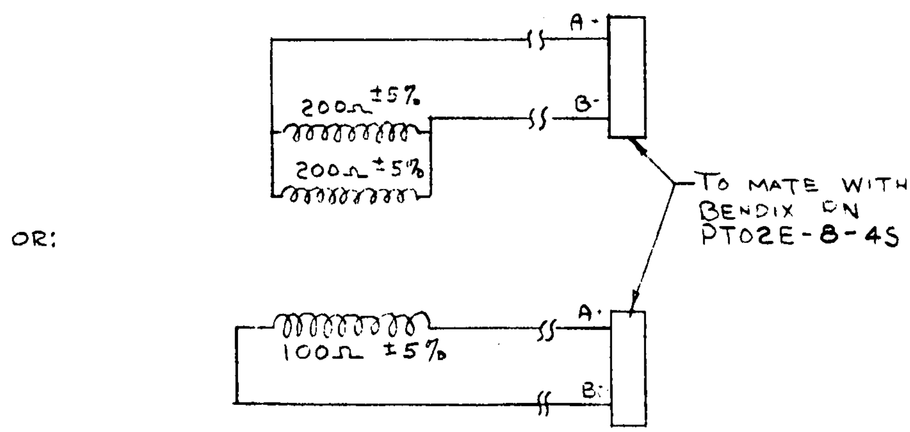
3812-100		11272	DS-659	A

ENGINEERING SPECIFICATION

TITLE	DATE
Torque Motor for Gimbal Valve Application	6-14-63

Input Characteristics	Value	
Rated current *	0.040 to 0.050	amps. max.
Input power at max. rated current	0.250	watts max.
Configuration (see Figure 1 & 2)	Two wire, parallel or series winding	
Coil resistance*	100	ohms equivalent (see figure)

* Most desirable combination is considered to be 100 ohms equivalent resistance and 50 ma rated current.



TORQUE MOTOR COIL ARRANGEMENT

FIGURE 1

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3812-100		11272	DS- 659	A

ENGINEERING SPECIFICATION

TITLE	DATE
Torque Motor for Gimbal Valve Application	8-14-63

Environmental	Value	
Operational temperature range	-250°F to +150°F	
Pressure	10^{-6}	mm. of Hg to sea level
Pressure differential across shaft seal	800	psi max.
Vibration		See Figure 3
Physical		
Size	See Figure 2	
Weight	Minimum	
Stroke stops	Permanent stops shall be set by vendor. Adjustable stops, with locking feature, shall be provided.	
Life		
Total service life at temperature	72 (130,000 cycles)	hours

Torque motor must be capable of performing within specifications after a 30-day soak at minus -250°F and 10^{-6} mm. Hg. This requirement shall also apply for an 8 hour soak at plus 150°F.

Operation cycles (±.015 in stroke)	10^6	cycles

Other Requirements

1. Magnetic gaps and coils must be sealed from working fluid at output.

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TITLE	DATE
Torque Motor for Gimbal Valve Application	8-14-63

Other Requirements

2. All threaded assemblies must be positively locked by safety wiring or other approved methods.

3. Mounting shall be with four holes through the torque motor base as shown in Figure 2.

4. The coil connections shall be made to a connector which mates with a Bendix type PT02E-8-4S or equivalent. All soldering connections shall be made in accordance with procedure MSFC-PROC-158A.

5. Current supplied to the coil from Pin A (+) to Pin B (-) shall cause the flapper output point to move toward the electrical connector.

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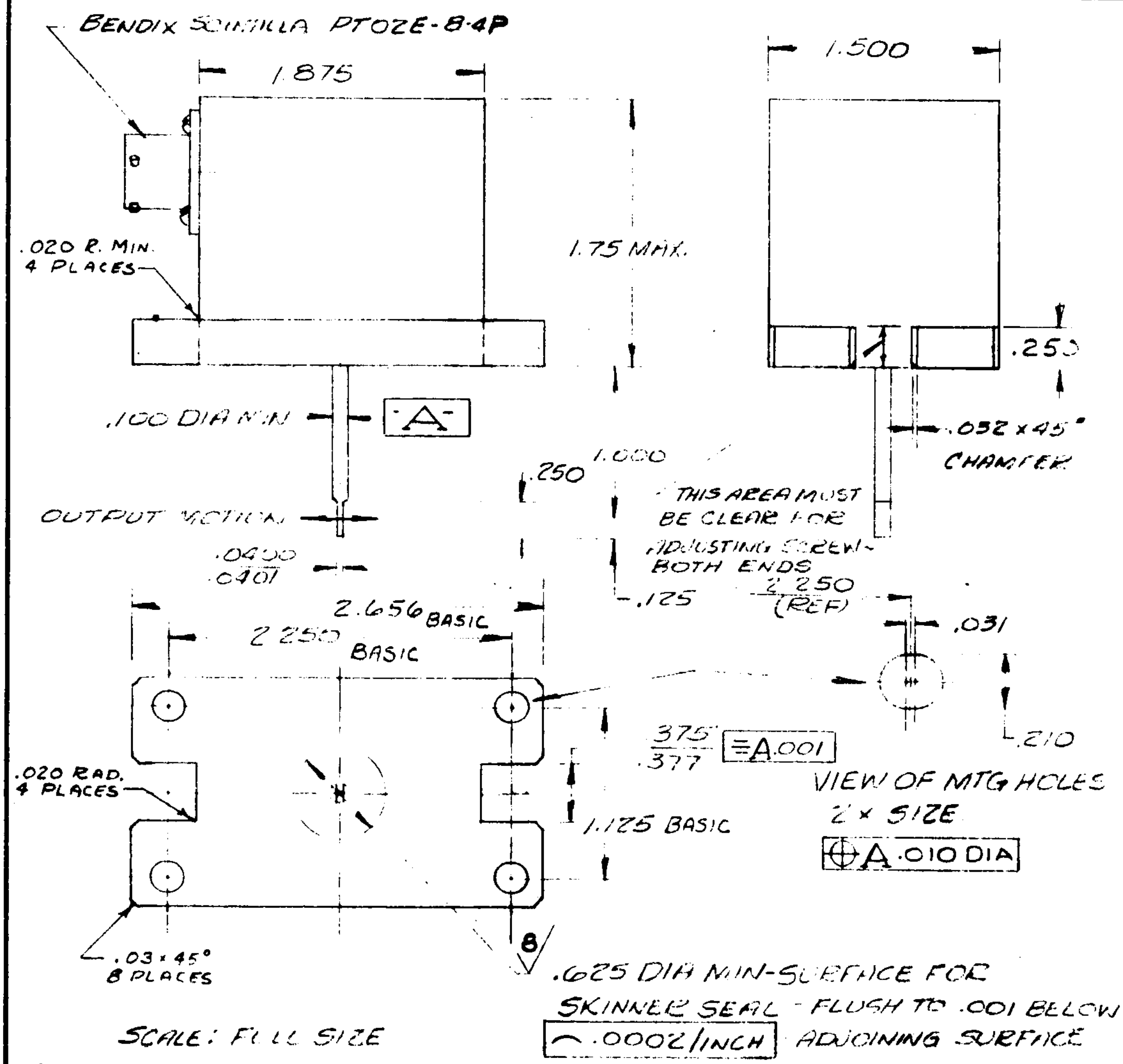
TITLE	DATE
Torque Motor for Gimbal Valve Application	8-14-63



BREAK SHARP EDGES 0.010 R MAX.
MATLS - CORROSION RESISTANT

FIGURE 2

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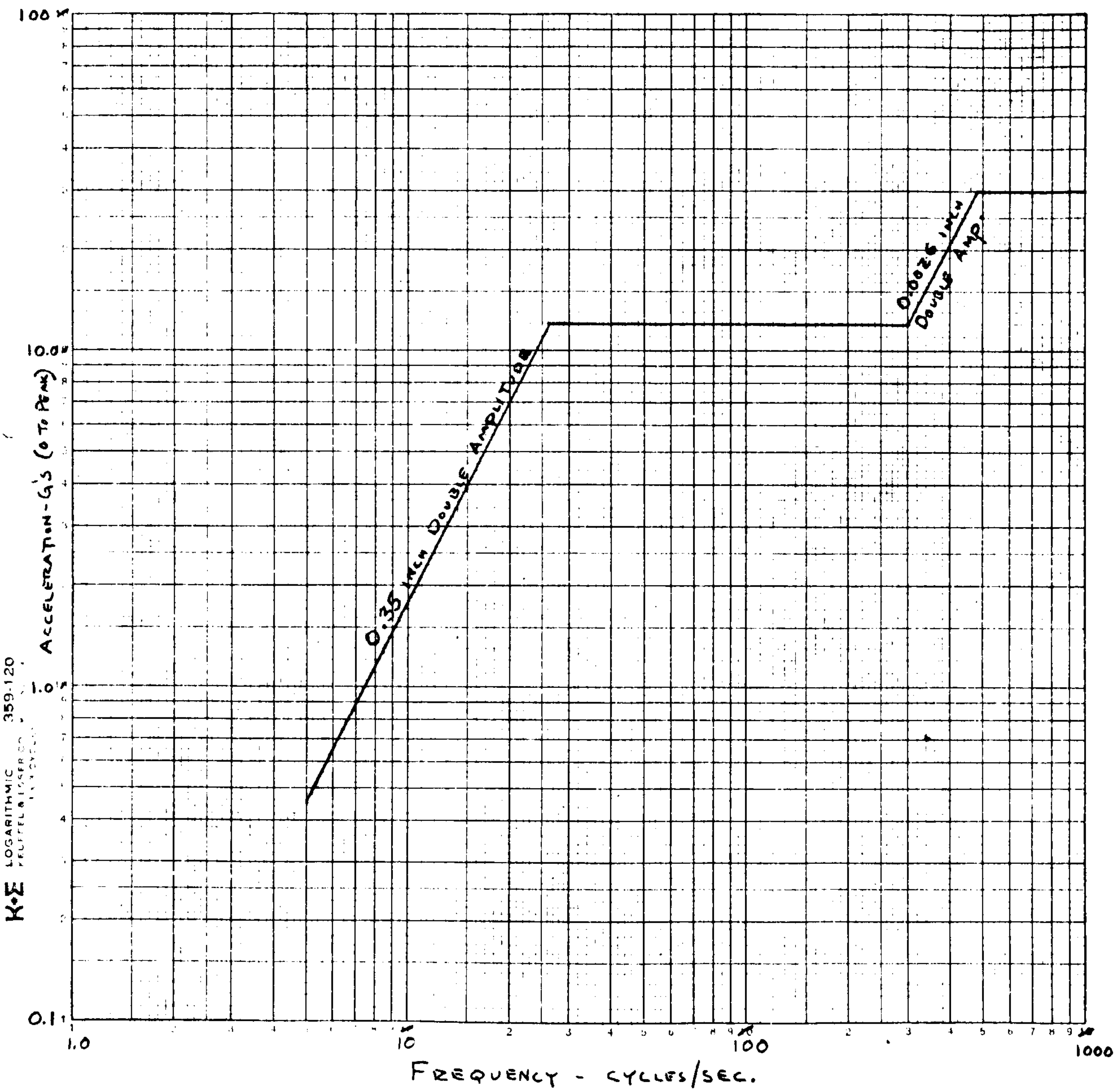
FIGURE 3



SINUSOIDAL SWEEP 4 MIN. PER OCTAVE - TEST TIME, ONE HOUR PER EACH OF THREE PRINCIPAL AXES.

Bendix

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
3812-302		11272	DS-668	A

ENGINEERING SPECIFICATION

TITLE Servo Valve No. 3812 (J-2 Actuator)

DATE January 7, 1964

Characteristic	Unit	Value
Gas Media		Hydrogen, Nitrogen
Operational Temperature Range	^{o}F	-250^{o} to $+150^{o}$
Lubrication Requirement		None
Rated Port Areas		
Supply	$in.^2$	0.0186
Exhaust	$in.^2$	0.0294
Valve Flow (Total)		
(a) Open Ports - N_2 @ 800 psig & $70^{o}F$	scfm	150
(b) Blocked Ports - N_2 @ 800 psig & $70^{o}F$	scfm	10
(c) 0.138 load orifice - N_2 @ 800 psig & $70^{o}F$	scfm	120
(d) 0.086 load orifice - N_2 @ 800 psig & $70^{o}F$	scfm	65
Operating Supply Pressure	psig	700 to 850
Rated Current	amps	± 0.050
Coil Resistance	ohms	100
Input Power	watts	0.250 maximum
Hysteresis	percent	<3
Null Shift over Temperature Range	percent	
Null Shift over Operating Pressure Range	percent	<2
Linearity	percent	4.5
Threshold	percent	0.5%
Spool Diameter	in.	0.625

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3812-302		11272	DS-668	A

ENGINEERING SPECIFICATION

TITLE	DATE
Servo Valve No. 3812 (J-2 Actuator)	January 7, 1964

Characteristic	Unit	Value
Spool Stroke	in.	±.015
Spool Lap Adjustment	in.	0±0.0001
Spool Diametral Clearance	in.	0.00025 to 0.0003
Volume Under Compression (Load Lines)	$in.^3$	
Nozzle Orifice Diameter	in.	0.031
Flapper Clearance per side	in.	0.002
Supply Port Proof Pressure	psig	1600
Return Port Proof Pressure	psig	800
Valve Body		440C stainless with black oxidize surface per MIL-13924A, class 2
Valve Spool		440C stainless with black oxidize surface per MIL-13924A, class 2
Electrical Connection		Bendix Scintilla PT02E-8-4P
Mating Connector		Bendix Scintilla PT02E-8-4S

NOTE: Current applied from pin A(+) to pin B(-) causes gas to flow from the load port nearest the electrical connector

Frequency Response		
At 800 psi & 70°F using nitrogen		
to ±3 db Amplitude	cps	50 (estimated)
90° phase lag	cps	102 (estimated)

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ENGINEERING SPECIFICATION

TITLE	DATE
Servo Valve No. 3812 (J-2 Actuator)	January 7, 1964

Characteristic	Unit	Value
At 800 psi & 70°F using hydrogen		
to ±3 db Amplitude	cps	
90° phase lag	cps	
Weight	lbs.	
Life: At -250°F	hours	72
	(cycles)	(130,000)
At 70°F (±0.015 stroke)	cycles	10^6
Valve Pressure Gain		
Blocked Ports	psi/amp	325,000
0.086 diameter load	psi/amp	38,250
0.138 diameter load	psi/amp	14,600
Open Ports	psi/amp	3,300

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	SPECIFICATION NO.	REV.
3312-301		DS-676	A

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

1. SCOPE

1.1 This specification establishes design, performance, and test requirements for a pneumatic rotary motor intended for use in an actuator control system for possible thrust vector control of the J-2 engine.

2. APPLICABLE DOCUMENTS - The following specifications, standards, drawings, and procedures constitute a part of this specification to the extent specified herein. Unless otherwise indicated, the revision and issue date shown shall apply.

SPECIFICATIONS

Military

MIL-C-6021	Castings, Classification and Inspection of, for Aeronautical Applications.
MIL-I-6181D	Interference Control Requirements, Aircraft Equipment.
MIL-S-7742A	Screw Threads, Standard, Optimum Selected Series, General Specification for.
MIL-T-9107	Test Reports, Preparation of.

STANDARDS

Military

MIL-STD-83	Dimensioning and Tolerancing.
MIL-STD-10A	Surface Roughness, Waviness and Lay.
MIL-STD-129	Marking for Shipment and Storage.
MS-21209	Insert, Screw Thread, Coarse and Fine, Screw Locking, Helical Coil, Cres.
MS-33540	Safety Wiring, General, Practice for.
MS-33586	Metal, Definition of Dissimilar.

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ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

DRAWINGS

MC-240 — Boss, Straight Thread, Standard Dimension for.

George C. Marshall Space Flight Center

MSFC-PROC-158A — Sodering of Electrical Connections (High Reliability), Procedure for.

OTHER PUBLICATIONS

Consolidated Classification Committee

Consolidated Freight Classification Rules

(Application for copies should be addressed to the Consolidated Classification Committee, 202 Union Station, Chicago 6, Illinois.)

American Trucking Association

National Motor Freight Classification Rules

(Application for copies should be addressed to the American Trucking Association, 1424 16th Street, NW., Washington, D. C.)

3. REQUIREMENTS

3.1 Motor - The motor shall be a pneumatic bi-directional, rotary output, positive displacement device capable of mounting to the power conversion transmission and utilize a servo-valve for the gas source.

3.1.1 Interchangeability - All like parts having the same part number shall be functionally and dimensionally interchangeable.

3.1.2 Metals - Metals used in fabricating this motor shall be the corrosion resistant type, or suitably treated to resist corrosion under all conditions specified herein, and those conditions likely to be met in storage or normal service.

3.1.2.1 Dissimilar metals - Where dissimilar metals are used, protection against electrolytic corrosion shall be provided, see Standard MS-33586.

PREPARED BY	CHECKED BY	APPROVED BY
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PROJECT NO.	SPECIFICATION NO.	REV.
3022-301	ES 675	A

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

3.1.3 Castings and forgings - All castings and forgings shall be of high grade quality, free from blow holes, cracks, porosity and sharp burrs. Castings shall be inspected and classified in accordance with Specification MIL-C-6021.

3.1.4 Screw threads - Screw threads shall be unified thread form class 3A or 3B, as applicable in accordance with Specification MIL-S7742A. Where tapped holes in aluminum alloy or magnesium are used, removable externally threaded steel inserts in accordance with Standard MS-21209 shall be used. This shall apply in those cases where frequent removal of the threaded part will be encountered in normal service or maintenance. Safety wiring shall be provided in accordance with Standard MS-33540.

3.1.5 Plumbing connections - All pluming connections shall be designed in accordance with Drawing MC-240.

3.2 Environment capability - The motor specified shall be capable of meeting all performance requirements under the following environmental conditions:

a. Temperature — -250 degrees Fahrenheit to 150 degrees Fahrenheit (F)

b. Pressure — 10^{-6} millimeters of mercury (mm Hg) to sea level

c. Vibration — 40 gravity (g)

3.3 Life & reliability - The motor shall be capable of operating for 72 hours under the conditions and requirements specified herein.

3.4 Motor operating conditions

3.4.1 Working medium - The pneumatic motor shall be designed to meet the performance requirements specified herein using cold gaseous hydrogen at minus 250 degrees F. and a nominal pressure of 800 plus 50 minus 100 pounds/square inch gage (psig) as a working medium. When specified, other mediums may be used provided that the gas temperature is selected to give a performance comparable to the operation with hydrogen at minus 250 degrees F.

3.4.2 Working pressures - The nominal working pressure shall be 800 psig. The motor shall be capable of withstanding, without permanent deformation or loss of performance, a proof supply pressure equal to 1600 psig and a proof return pressure of 600 psig. Design burst pressure shall be 3200 psig for supply and 1000 psig for return.

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3812-301		DS-676	A

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

3.5 Motor response

3.5.1 Open loop frequency Response - The motor and valve are approximated by a second order system with a natural frequency of 24 cps (no load) with a damping factor of 1.0.

3.6 Design and construction

3.6.1 Size & weight - The motor shall be designed to fit within the available space noted on figure 1. Maximum weight of the system, shall not exceed 5.5 pounds.

3.6.2 Design criteria - The motor shall be designed and constructed to withstand the strains, shocks, vibrations, and other conditions incident to shipping, storage, installation, handling, and operational service as defined in this specification.

3.6.3 Lubrication - Self lubrication shall be used in all necessary areas. The lubrication shall be adequate for those environmental conditions of paragraph 3.2. Lubrication shall be adequate for a 72 hour duty cycle or 130,000 operating cycles at 1/2 rated load, 1/2 stroke and 0.5 cps Seal changes will be permitted at 50,000 cycle intervals. The motor shall be capable of performing within the requirements of this specification after a 30 day soak at minus 250 degrees F and 10^{-6} mm Hg. This requirement shall also apply for an 8 hour soak at plus 150 degrees F.

3.6.4 Quiescent Flow - The quiescent flow shall be no greater than 0.5 pounds per minute with the servo valve at null and with supply pressure connected.

3.6.5 Motor Output - The steady state output of the motor shall be an angular velocity proportional to the input signal.

3.7 Motor Type - The motor shall consist of all parts necessary to convert the stored energy of hydrogen gas into a rotary output. The motor shall be a positive displacement, bi-directional continuous rotary device.

3.7.1 Motor Torque - The motor shall be designed so the maximum output torque shall not exceed 220 inch-pounds of torque.

3.7.2 Motor velocity and acceleration - With a simulated inertia load of 6.6×10^{-3} pound-inch seconds^2, the motor shall be capable of producing an angular velocity of 3500 RPM and an acceleration of 4.03×10^3 radians per second^2. The maximum no load velocity of the motor shall not exceed 7000 RPM with a pressure drop

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TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

See RLD Layout No. F-215396

Figure 1 Envelope

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E. Łozniecki		

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ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

See RLD Layout No. E-215396

Figure 2 Porting

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ENGINEERING SPECIFICATION

of 600 psig across the motor.

3.7.3 Rated load - Rated load of the motor shall be 47 inch pounds of frictional torque and an inertial load consisting of the transmission and load of 6.0×10^{-3} pound-inch-seconds^2. The motor inertia shall not exceed 1.5×10^{-3} pound-inch-seconds^2.

3.7.4 Attach Points - The motor shall have mounting pads designed for given seal tight joints between the servo-valve and the transmission and giving adequate structually rigidity and strength for proof and burst testing when assembled to the system.

3.7.5 Threshold - The increment of valve current to change the direction of an unloaded motor shall not exceed, 3.5 ma with 800 psig applied to servo-valve.

4. Quality Assurance Provisions

4.1 Testing - Unless otherwise specified, tests may be conducted with a gaseous medium at a temperature which will provide performance comparable to hydrogen at minus 250 degrees F. System pressure shall be 800 plus 50 minus 100 psig. Ambient pressure shall be 20 to 30 inches Hg.

4.1.1 Properly Connected - When the motor is referred to as properly connected it shall mean that the system is loaded as in paragraph 3.7.2.

4.1.2 Gas Consumption - With the system properly connected an input shall be applied which is equivalent to ± 17.5 revolutions of the motor. This signal shall be varied sinusoidal at 0.3 cps. The total gas consumption of the motor shall not be greater than 3 pounds per minute.

4.1.3 Velocity - With the motor-valve combination connected open loop, rated current, shall be applied to the valve. The simulated load velocity shall be 3500 RPM ± 350 RPM. The test shall be repeated for 75, 50, 25, and 10% of rated current. Load velocity as determined at each given percent of rated current shall be in accordance with figure 3 .

4.1.4 Acceleration - Rated current shall be applied to the servo valve as a step input. The maximum output of the motor shall be not less than 4.63×10^3 rad/sec^2. This test shall be repeated for 75, 50, 25, and 10 percent of rated current.

4.1.5 Flow plot - A flow plot of the servo valve-motor combination under no load and rated load conditions shall be made and the data recorded. The plot shall show valve flow versus current. All curves shall include data for 150 percent of

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ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964

rated current.

4.1.6 Pressure plot - A pressure plot of the motor recovery pressure under no load and rated load conditions shall be made and the data recorded. The plot shall show ΔP as a function of current. All curves shall include data for 150 percent of rated current.

4.1.7 Null bias measurement - With the valve properly connected, the null bias current shall be no greater than 0.50 ma.

4.1.8 Threshold - The threshold of the motor-valve combination as defined, shall be determined and shall be no greater than 3.5 ma.

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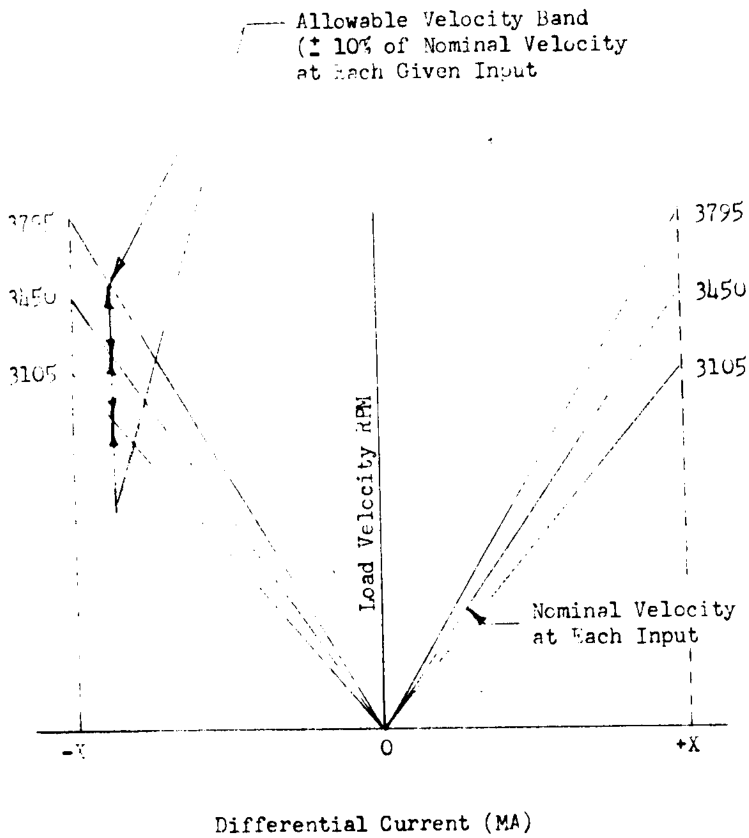
TITLE	DATE
Pneumatic Rotary Motor	January 8, 1964



X denotes rate current

Figure 3 -VELOCITY LIMITS

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APPENDIX C

ANALOG SIMULATION STUDIES OF J-2 ENGINE CONTROL - PNEUMATIC ROTARY VANE ACTUATOR

This Appendix summarizes the analog simulation studies of a J-2 engine control system using a pneumatic rotary vane actuator and a two-stage jet pipe valve. The system response is evaluated under the effects of the revised structural resonance and coulomb friction load on the engine. Particular attention is focused on the engine dynamic performance. The results show that with minor changes in the actuator parameters the specifications can be met.

The block diagram, Figure C-1, is set up such that it closely resembles the physical system, especially the locations of the saturation limits. The amplifier used is the one specified, with 0.7 damping and 300 rad/sec natural frequency. The torque motor gain K_{TM}, is 2×10^{-3} in/ma. The servo valve pilot stage time constant, τ_v, is set at 1/340 seconds or the first order break frequency of 54 cps. The mass spring system of the servo valve is neglected because its undamped natural frequency is over 300 cps and there is very low damping in the valve.

Motor pressure feedback, if needed, is summed at the jet pipe. Just as in the physical case, the pressure acting on some bellows area becomes a force which works against a spring resulting in a jet pipe position proportional to the pressure. The loop closure around the jet pipe position saturation limit results from writing the valve flow and force summation at the jet pipe. This is shown in detail, in Appendix C-1. The two partial terms, $\partial p/\partial y$ and $\partial p/\partial Q$, representing the characteristic curves of the valve and motor, are reasonably linear and, thus, are represented by constants. τ_1 represents the time constant of the motor or the effect of compressibility of gases on the time required to pressurize the motor. τ_1 is set at 0.007 second. The motor pressure, acting on the displacement of the motor times the transmission ratio produces a force on the engine load. Minor loops are added to account for the transmission efficiency of 80 percent and the actuator coulomb friction. A position loop is closed around the actuator. The feedback potentiometer has a gain, K_f, of 20 volts/in.

Before the problem is put on the computer, a brief analysis is made so that it can be used as a guide. The amplifier dynamics and the valve lag are neglected and a transfer function written for the actuator position to command signal.

$$\frac{\beta p}{y} = \frac{\frac{1}{K_f}}{\frac{J_m \tau_1 R}{d_m K_f K_a K_{Tm} \frac{\partial p_m}{\partial y}} S^3 + \frac{J_m R}{\frac{\partial p_m}{\partial y} K_f K_a K_{Tm} d_m} \left[1 \pm \left(\frac{1 + \tau_2 S}{1 + \tau_3 S}\right) \frac{\partial p_m}{\partial y} K_p\right] S^2 + \frac{\frac{\partial p_m}{\partial Q} d_m R}{\frac{\partial p_m}{\partial y} K_f K_a K_{Tm}} S + 1}$$

It can be seen that

$$\omega_{ns}^2 = \frac{d_m^2 \frac{\partial p_m}{\partial Q}}{J_m \tau_1}$$

$$\omega_{ns} = 24 \text{ cps}$$

$$\frac{\alpha}{\beta} = \frac{\omega_{ns}^2 J_m R}{\frac{\partial p_m}{\partial y} K_f K_a K_{Tm} d_m} \left[1 \pm \left(\frac{1 + \tau_2 S}{1 + \tau_3 S}\right) \frac{\partial p_m}{\partial y} K_p\right]$$

$$\beta = \frac{K_f K_a K_{Tm} \frac{\partial p_m}{\partial y}}{\frac{\partial p_m}{\partial Q} d_m R \omega_{ns}} = 0.01715 K_a$$

It is quite obvious therefore that positive pressure feedback decreases α and negative pressure feedback increases α. Variation in forward loop gain, which in this case, is accomplished by varying K_a, results in a proportional change in β. α can be interpreted as the damping ratio. However, discretion must be used in changing α since there is a point of diminishing return depending on what the value of β is. The ω_{ns} of the actuator can be used to check portions of the computer set up. By disengaging the engine and friction load from the actuator, the servo actuator will oscillate at its undamped natural frequency when K_a is set sufficiently high. The oscillation is observed to be 32 cps. The calculated undamped natural frequency of the actuator is 24 cps. However, in the computer data, the amplifier and valve phase lags which accounts for about 30 degree at approximately 20 cps are also included. These lags theoretically reduce the actuator resonant frequency to 16. 5 cps. The mass spring system of the engine is checked in the same manner. Its natural frequency is 9. 2 cps.

The dynamic response requirements are listed as follows:

1. Transient response - the transient response of the system shall have a rise time, τ_r, of less than 0. 419 second and settling time, τ_s, of less than 0. 35. The overshoot shall be less than 30 percent (Figure C-2).

2. Closed loop frequency response - the closed loop frequency response of the system shall have no greater than 20 degrees phase lag at 1 cps and a peak amplitude ratio of not greater than 1. 5 (+3. 5 db), system bandwidth (-3 db) shall be 8 cps or greater.

An engine friction load of 730 pounds is used to tune up the control system. It has been determined that with minor changes in the $\vartheta p/\vartheta Q$ term from 1. 5 to 1. 7 and τ_1 from 0. 01 second to 0. 007 second and K_a set at 14, all the dynamic response requirements can be met. Since the vane motor used here, unlike the gear motor, is not overdamped, no motor pressure feedback is necessary. The gain adjustment setting K_a, is determined by observing the frequency response peak and bandwidth since experience have shown that the transient response requirement can generally be met. The engine and actuator frequency response is shown in Figure C-3 and plotted in Figure C-4. The engine response peaks to +3 db at 5. 6 cps. The bandwidth is at 8 cps. At 1 cps, the phase lag is 18 degrees. The closed loop transient response for 6, 10, 50, 75, 100 percent amplitude steps are shown in Figure C-5. The time scale is changed to check the rise time and settling time (Figure C-6). The rise time and settling time are 0. 041 second and 0. 212 second respectively and the overshoot is 18 percent. Static stiffness at the actuator with various loading is shown in Figure C-7. With 20, 000 pound load on the engine, the measured stiffness is 1. 35 x 10^6 lb/in. The dynamic stiffness response is shown in Figure C-8. A sinusoidal 10, 000 pound load disturbance is applied and the actuator and engine position recorded. The dynamic stiffness of the servo motor and engine is then plotted in Figure C-9. The open loop velocity and acceleration data are also taken. Figure C-10 shows that the actuator saturation velocity is approximately 4 in/sec. and the maximum acceleration is 40 in/sec^2.

The engine position frequency response with 7, 300 pound friction load is also shown in Figures C-3 and C-11. The bandwidth is reduced to 5 cps and the phase lag at 1 cps is over 50 degrees. It should be noted here that compensation network can be added to increase the bandwidth to 8 cps and reduce the phase lag at 1 cps. A simple lag-lead network needed here can even be achieved by all mechanical means as shown in Appendix C-2. If the dynamic response is tuned with 7, 300 pound friction, however, it is very likely that should the friction be reduced in any manner, the response characteristic will change considerably. Since the actual engine friction load can be in the range of 730 pounds, the parameters determined would serve as a guide in the preliminary design of the actuator system.

APPENDIX C-1

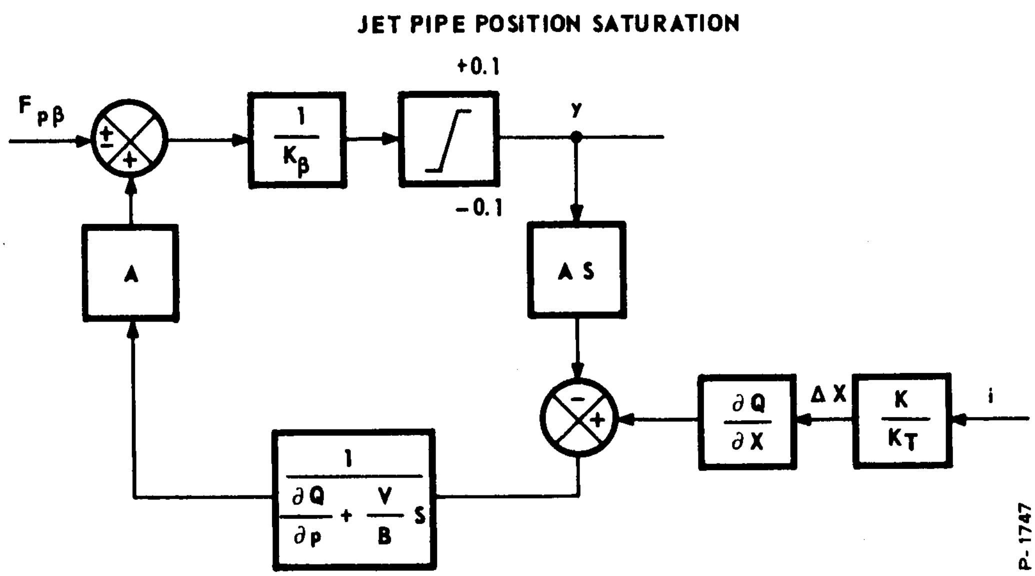
The pilot stage and the jet pipe value is shown schematically in Figure 12. The flow equation after linearization is:

$$\frac{\partial Q}{\partial x} \Delta x = A\ S\ y + \left(\frac{\partial Q}{\partial p} + \frac{V}{B} S \right) \Delta\ p \qquad (1)$$

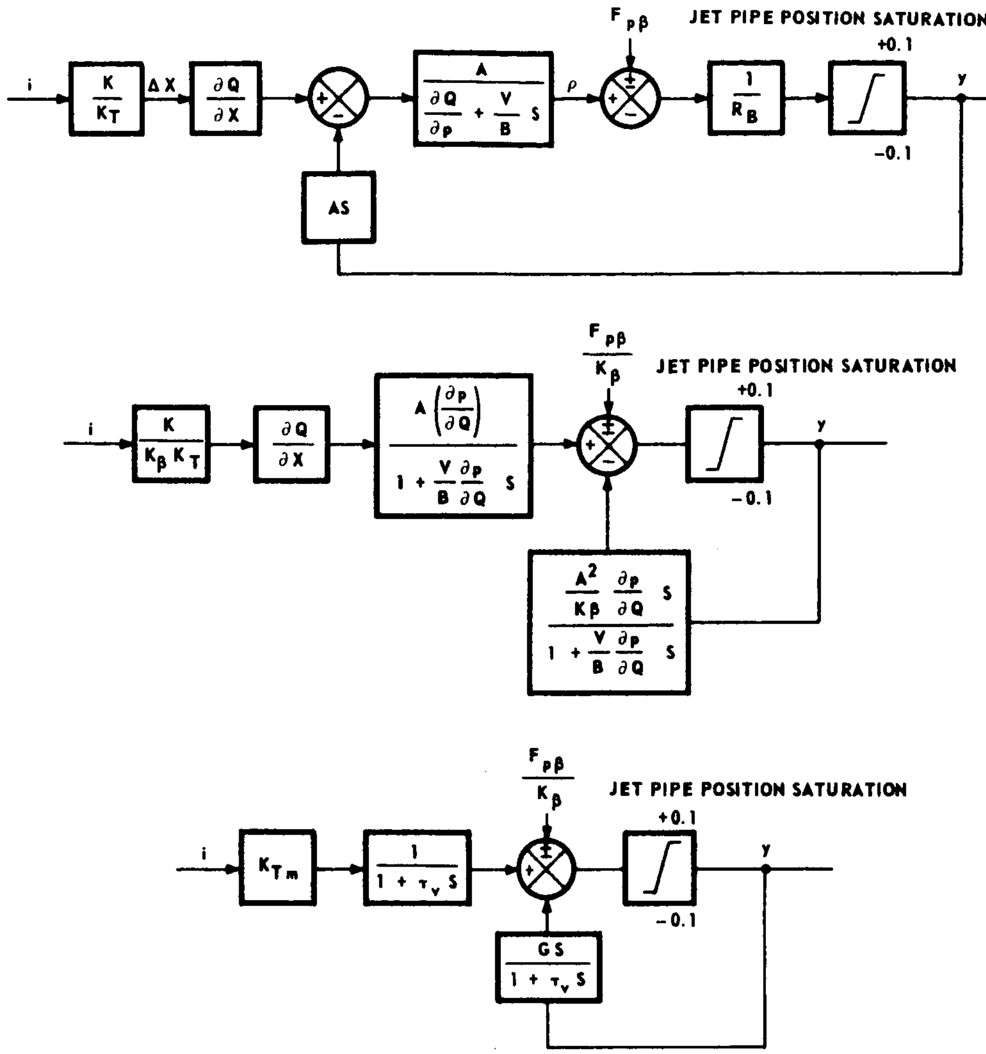
The pressure force, acting on the bellows area A, together with the positive or negative pressure feedback force, work against a spring and positions the jet pipe. Thus,

$$\frac{\pm F_{p\beta} + \Delta pA}{k_{\beta}} = y$$

The block diagram can thus be set up from Equations (1) and (2)



This can be modified as follows.



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Let

$$K_{Tm} = \frac{K}{K_\beta K_T} \frac{\partial Q}{\partial x} A \frac{\partial p}{\partial Q} = 2 \times 10^{-3} \text{ ma/in}$$

$$\tau_V = \frac{V}{B} \frac{\partial p}{\partial Q} = \frac{1}{340} \text{ sec}$$

$$G = \frac{A^2}{k_\beta} \frac{\partial p}{\partial Q} = \frac{1}{3400} \text{ sec}$$

where

K = Torque motor force gain, lb/ma

K_T = Torque motor displacement gain, in/lb

V = Volume under compression, in.^{3}

B = bulk modulus

APPENDIX C-2

ALL MECHANICAL LAG LEAD NET WORK

A lag lead net work can be achieved by a simple orifice and a bellow as shown in Figure 13.

$$A_\beta S x + c A_o R \sqrt{T} \frac{p_m}{p} = \frac{V}{kp} S p$$

where

A_β = Compensation bellow mean effective area.

x = Compensation bellow stroke.

c = $c_d \times c_2$

= Coefficient of orifice discharge x c_2 of the fluid.

A_o = Orifice area

R = Gas constant

T = Gas temperature

V = Volume under compression downstream of orifice

k = Specific heat ratio.

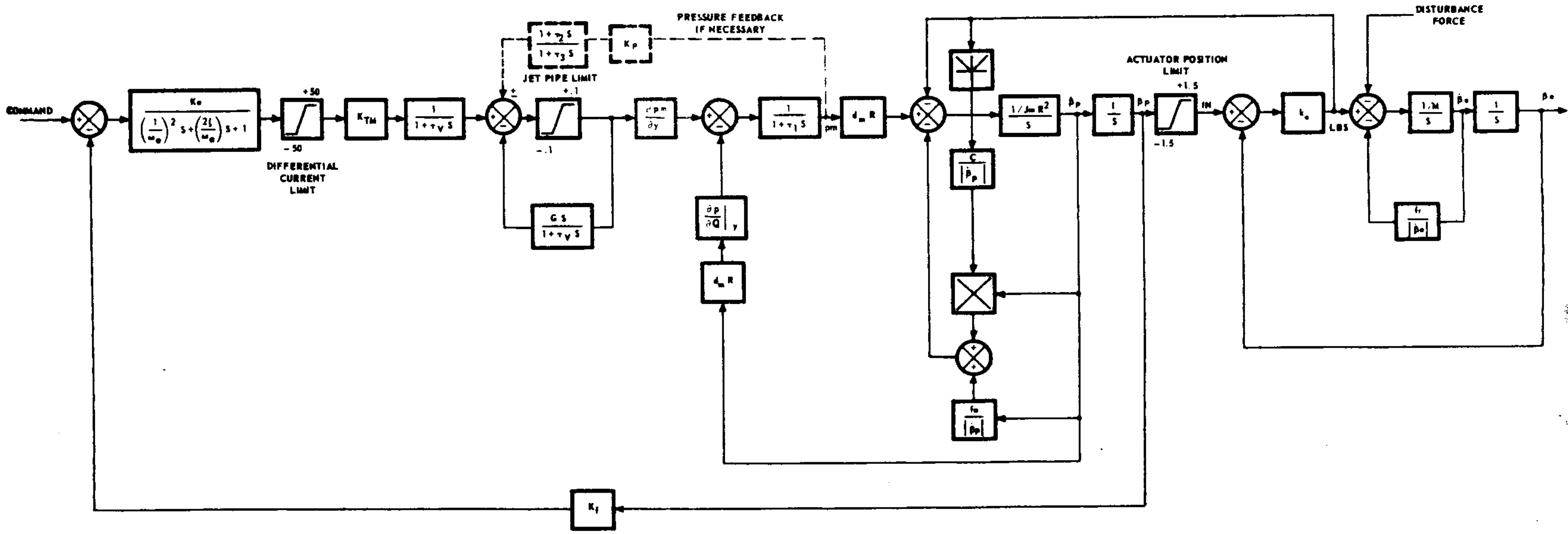
Linearizing and substituting $\quad x = \frac{p_m A_o}{k_\beta}$

$$\frac{p_m A_\beta^2}{k_\beta} S + \frac{c A_o R \sqrt{T} p_m}{p_o} = \frac{p_{mo} c A_o R \sqrt{T}}{P_o^2} p + \frac{V}{k p_o} s p$$

$$\frac{p}{p_m} = \frac{\dfrac{c A_o R \sqrt{T}}{p_o} + \dfrac{A_\beta^2}{k_\beta} s}{\dfrac{p_{mo} c A_o R \sqrt{T}}{p_o^2} + \dfrac{V}{k p_o} s}$$

$$= \left(\frac{1 + \dfrac{A_\beta^2 \, p_o}{k_\beta c A_o R \sqrt{T}} s}{1 + \dfrac{V p_o}{k c A_o R \sqrt{T}} s} \right) \frac{p_{mo}}{p_o}$$

Thus, the lag lead corner frequencies can be adjusted to the proper values by varying the parameters.



K_a = AMPLIFIER STATIC GAIN

ω_a = AMPLIFIER NATURAL FREQUENCY 300 RAD/SEC

K_{TM} = TORQUE MOTOR GAIN = 2×10^{-3} IN/ma

τ_V = SERVO VALVE TIME CONSTANT = $\frac{1}{340}$ SEC

G = VALVE FEED BACK GAIN = $\frac{1}{3400}$ SEC

$\frac{\partial pm}{\partial y}$ = VALVE POWER STAGE GAIN = 7000 PSI/IN

$\frac{\partial pm}{\partial Q}$ = VALVE PRESSURE FLOW GAIN = 1.7 PSI/SEC

d_m = MOTOR DISPLACEMENT = .53 IN^3/RAD

R = TRANSMISSION RATIO = 173 RAD/IN (ROTARY TO LINEAR)

K_f = FEED BACK POT GAIN 20 VOLT/IN

τ_1 = MOTOR COMPRESSIBLE FLOW = .007 SEC TIME CONSTANT

K_p = MOTOR PRESSURE FEED BACK GAIN

J_m = MOTOR INERTIA = .003 IN - LB - SEC^2

C = TRANSMISSION EFFICIENCY = 0.8

f_m = MOTOR COULOMB FRICTION = 850 LBS.

k_e = EQUIVALENT SPRING RATE OF = 3.91×10^5 LB/IN ENGINE & STRUCTURE

M = EQUIVALENT MASS OF ENGINE = 116 $\frac{LB - SEC^2}{IN}$ TO THE ACTUATOR

f_r = COULOMB FRICTION IN THE LOAD = 730 LB

Figure C-1. Engine Control System Block Diagram Used in Computer Simulation

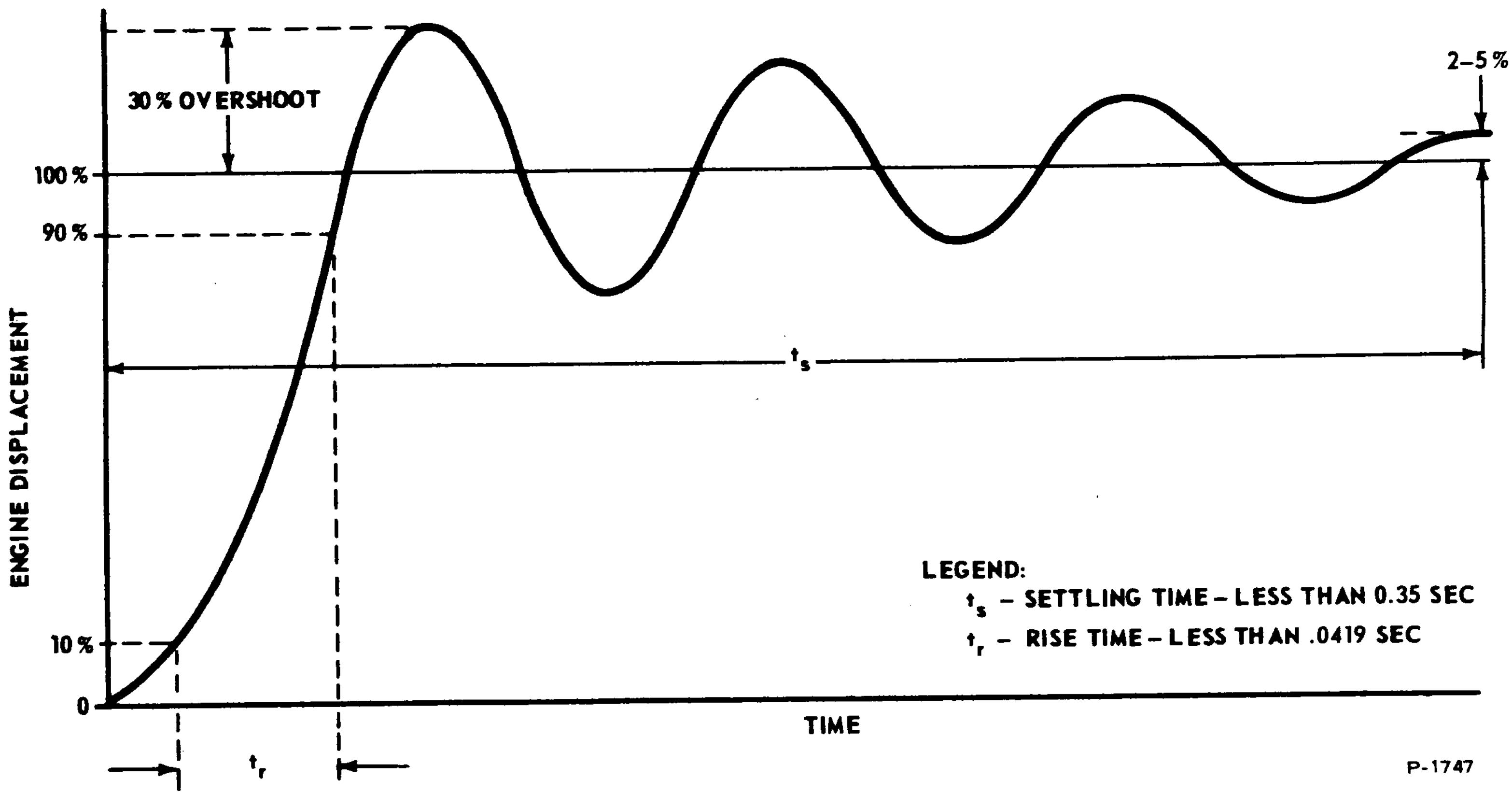


Figure C-2. Transient Response Requirement

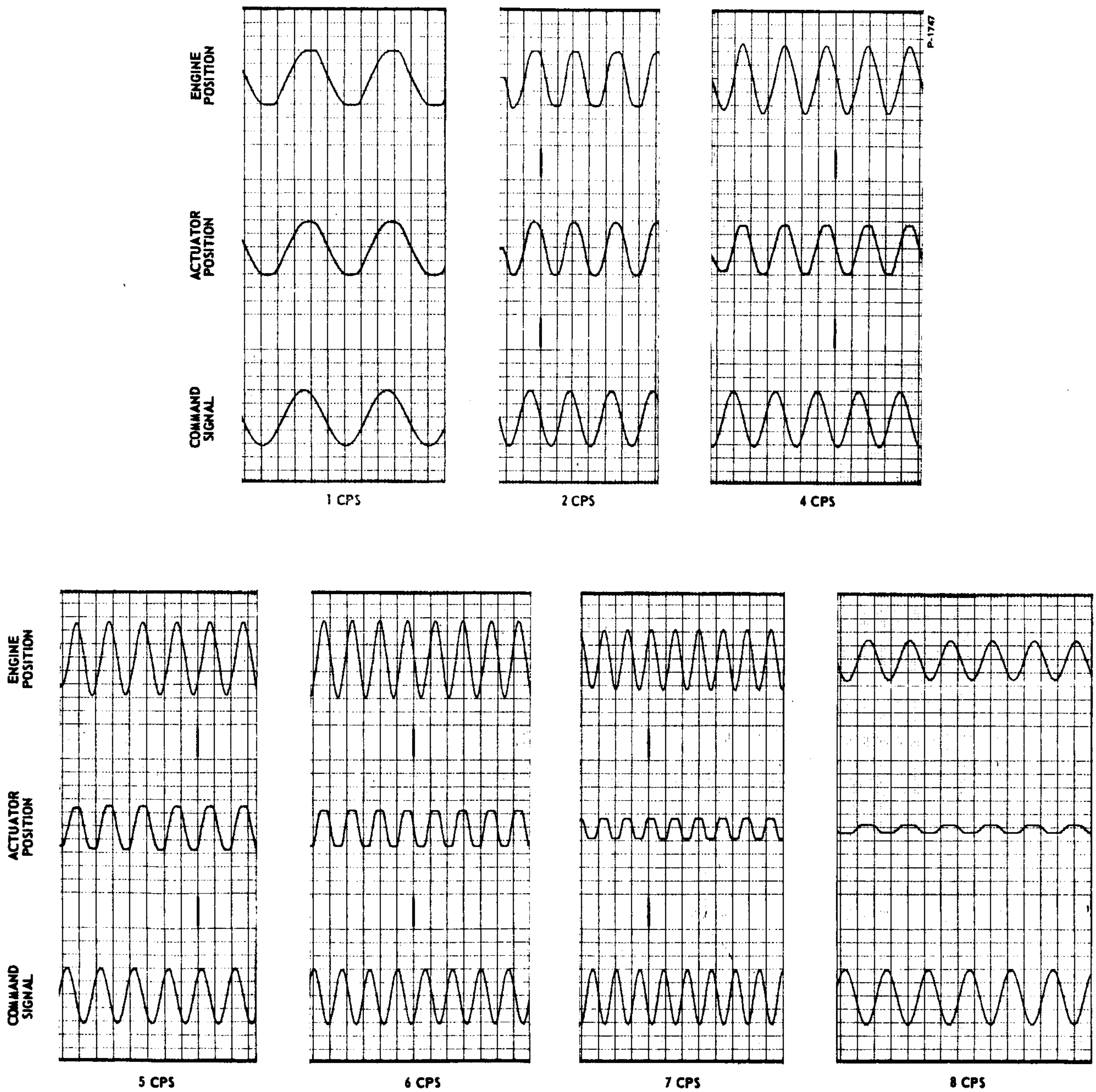


Figure C-3. Actuator and Engine Frequency Response (730 LB Friction Load)

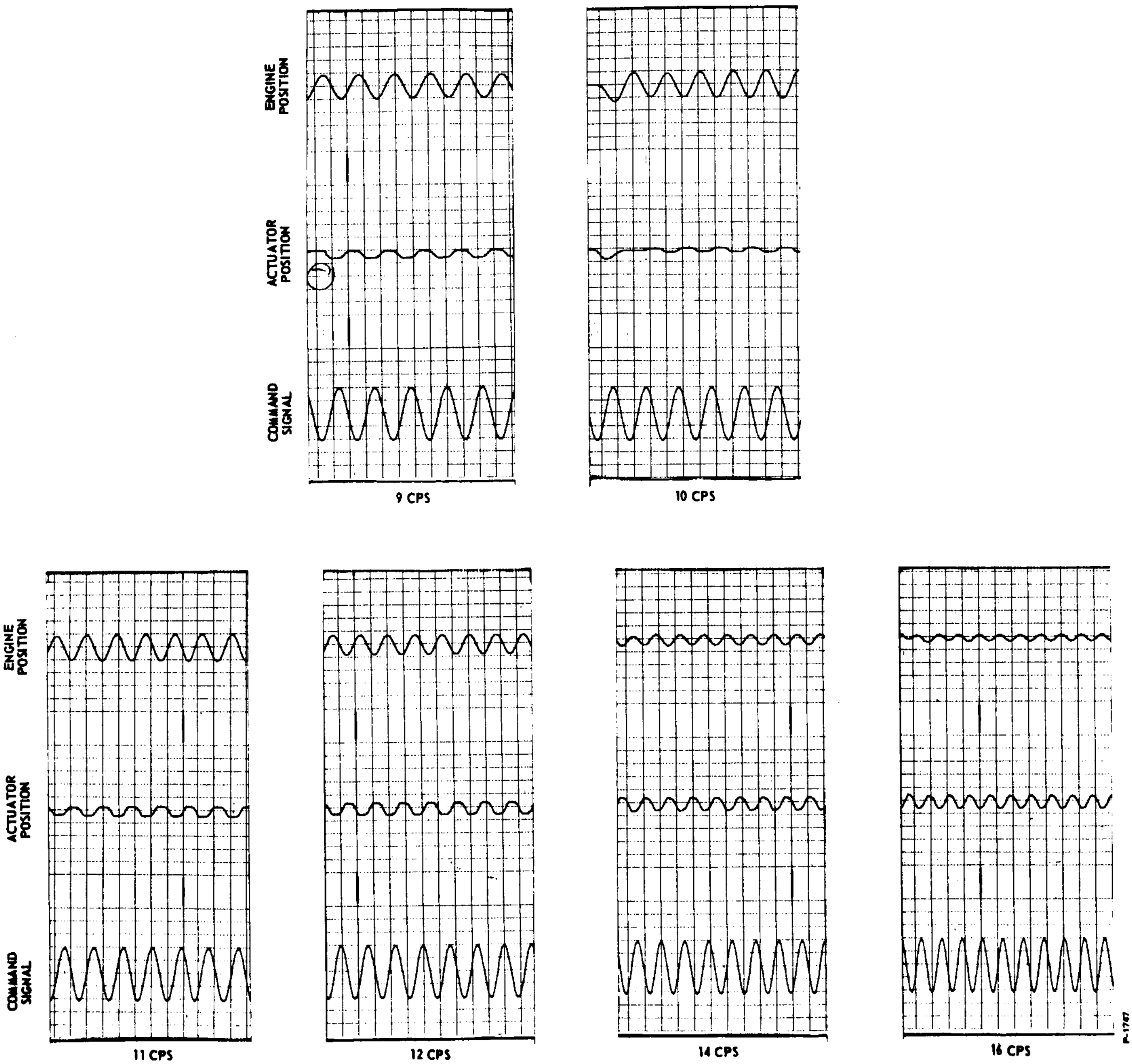
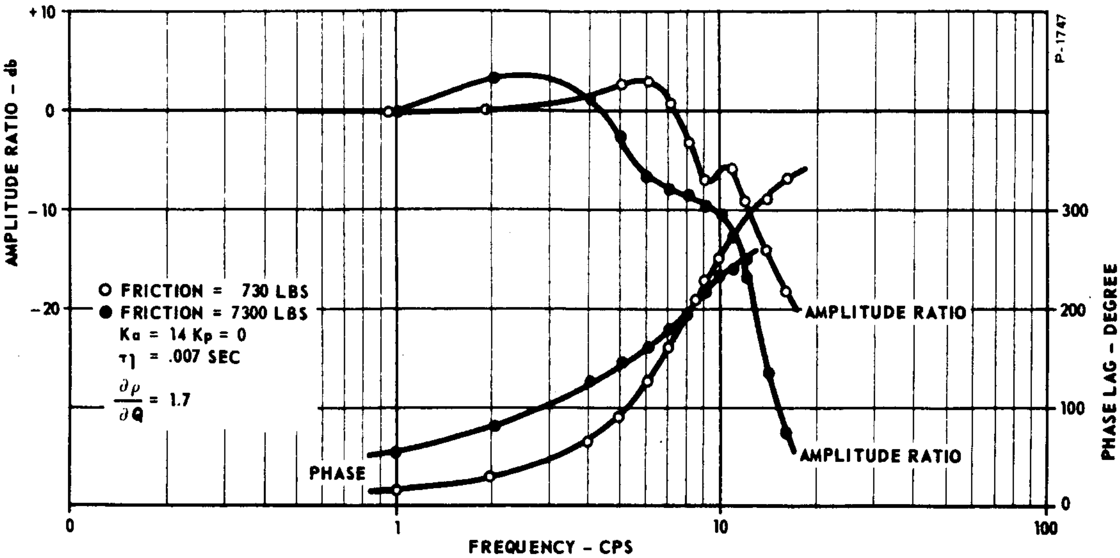
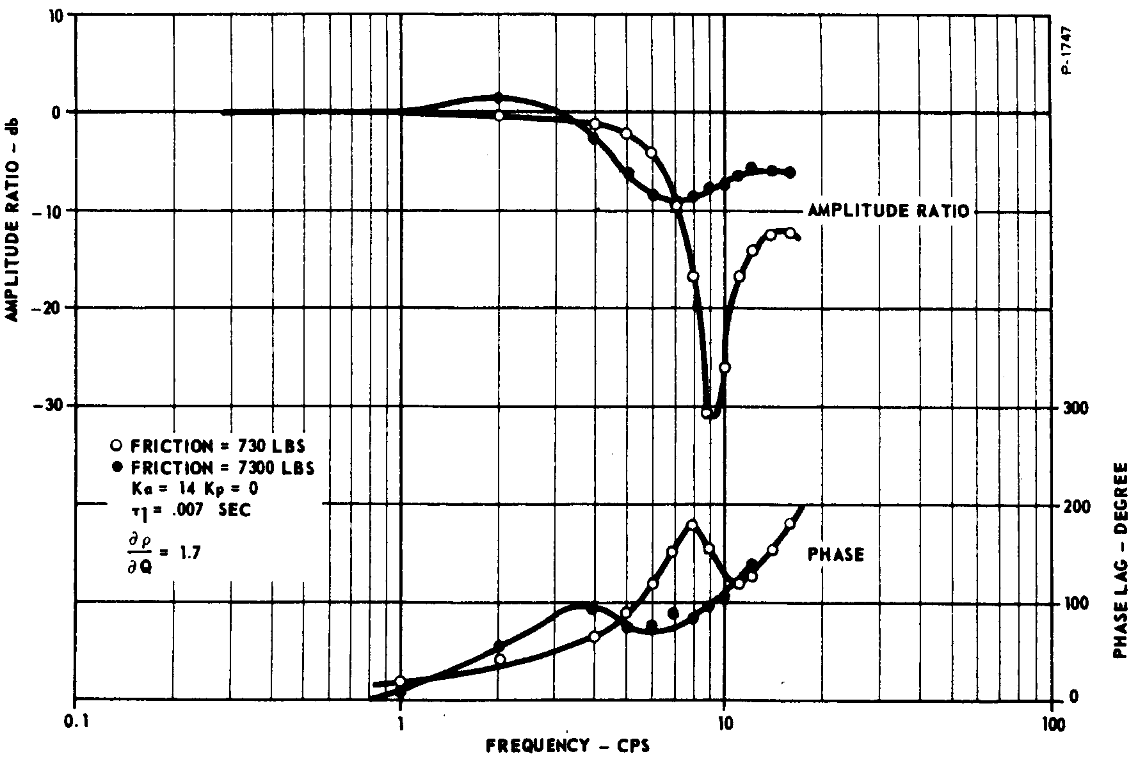


Figure C-3 (Cont'd). Actuator and Engine Frequency Response
(730 LB Friction Load)



4a ENGINE FREQUENCY RESPONSE



4b ACTUATOR FREQUENCY RESPONSE

Figure C-4. Actuator and Engine Frequency Response

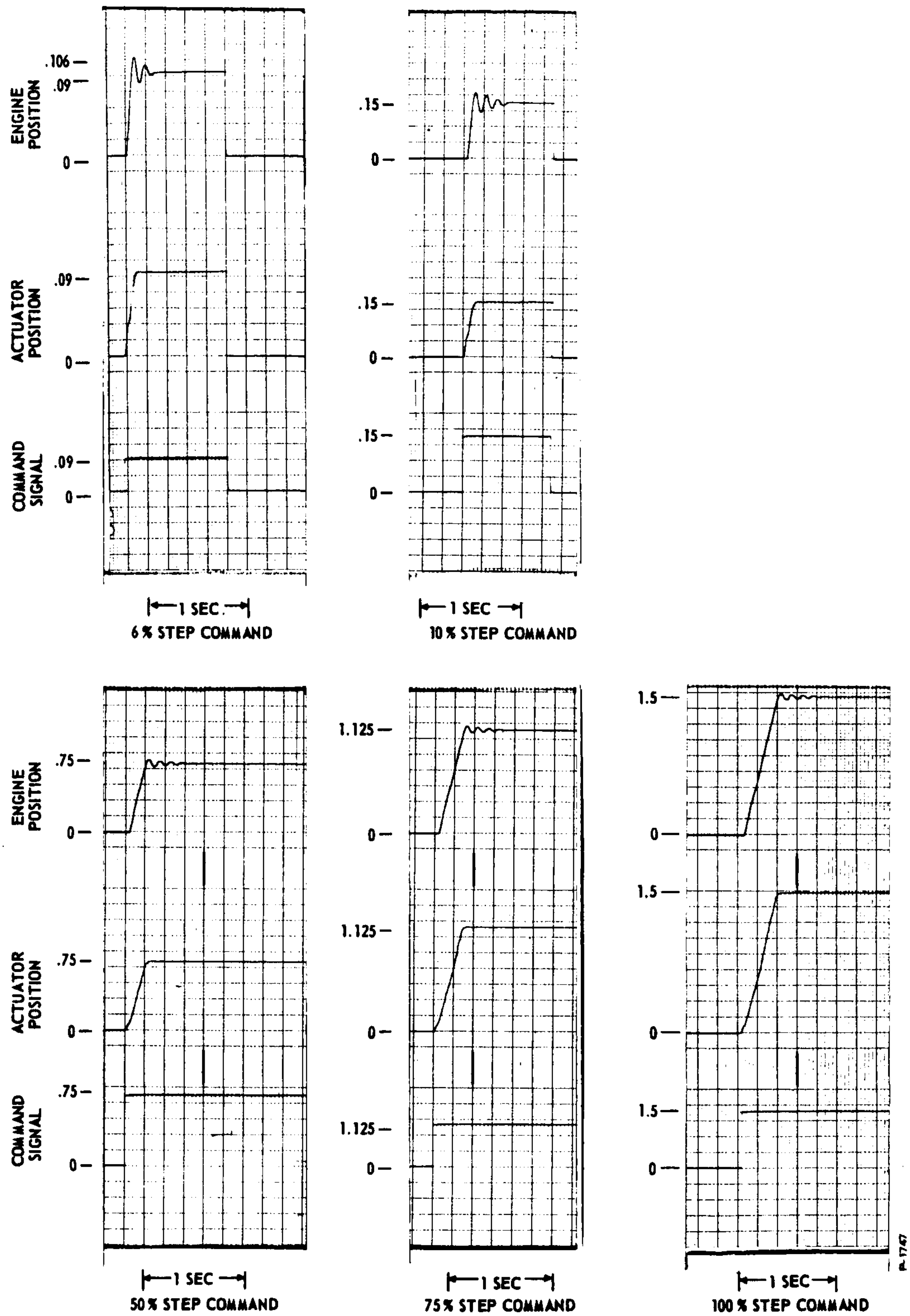


Figure C-5. Actuator and Engine Transient Response

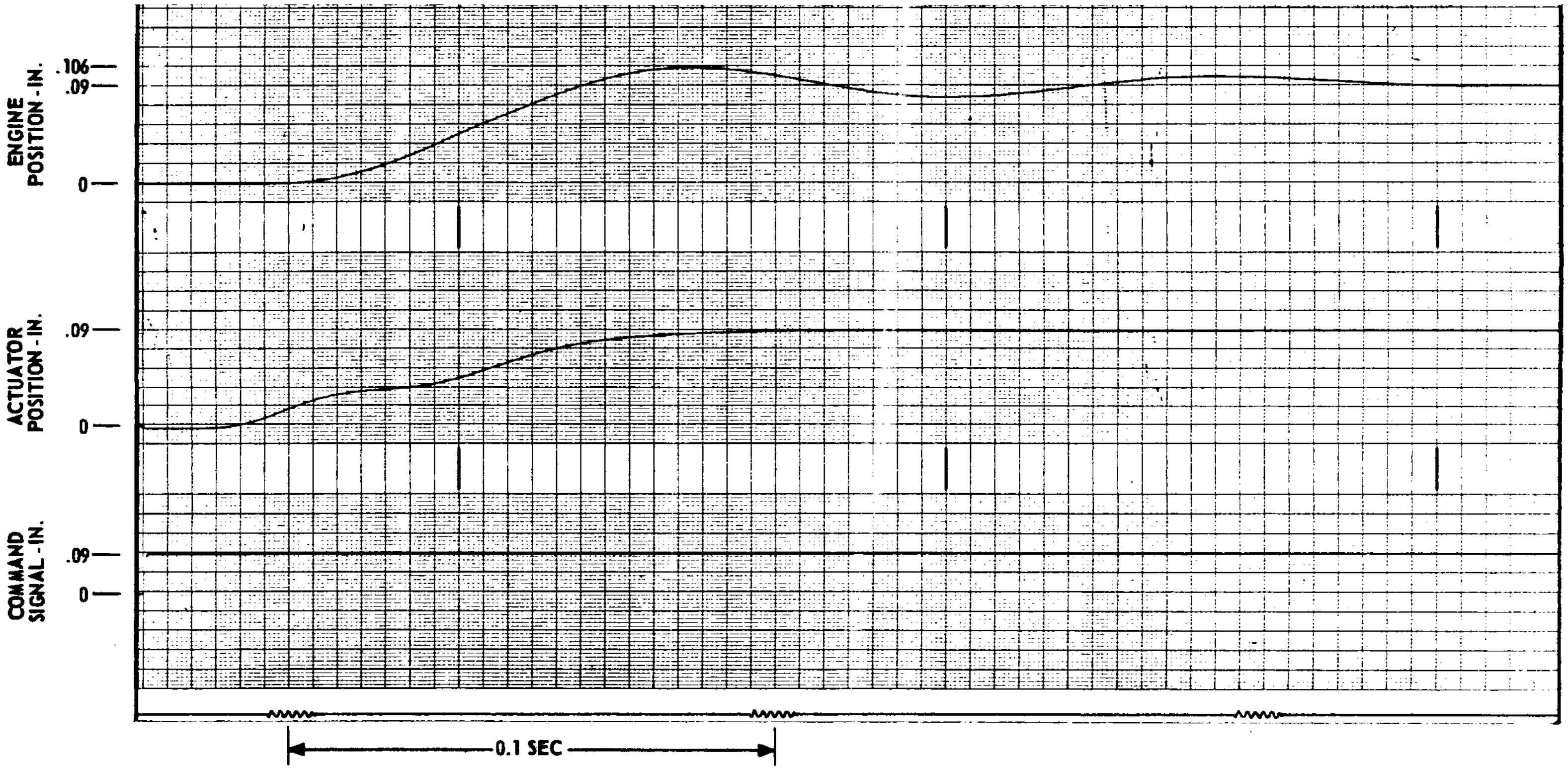


Figure C-6. Actuator and Engine Transient Response

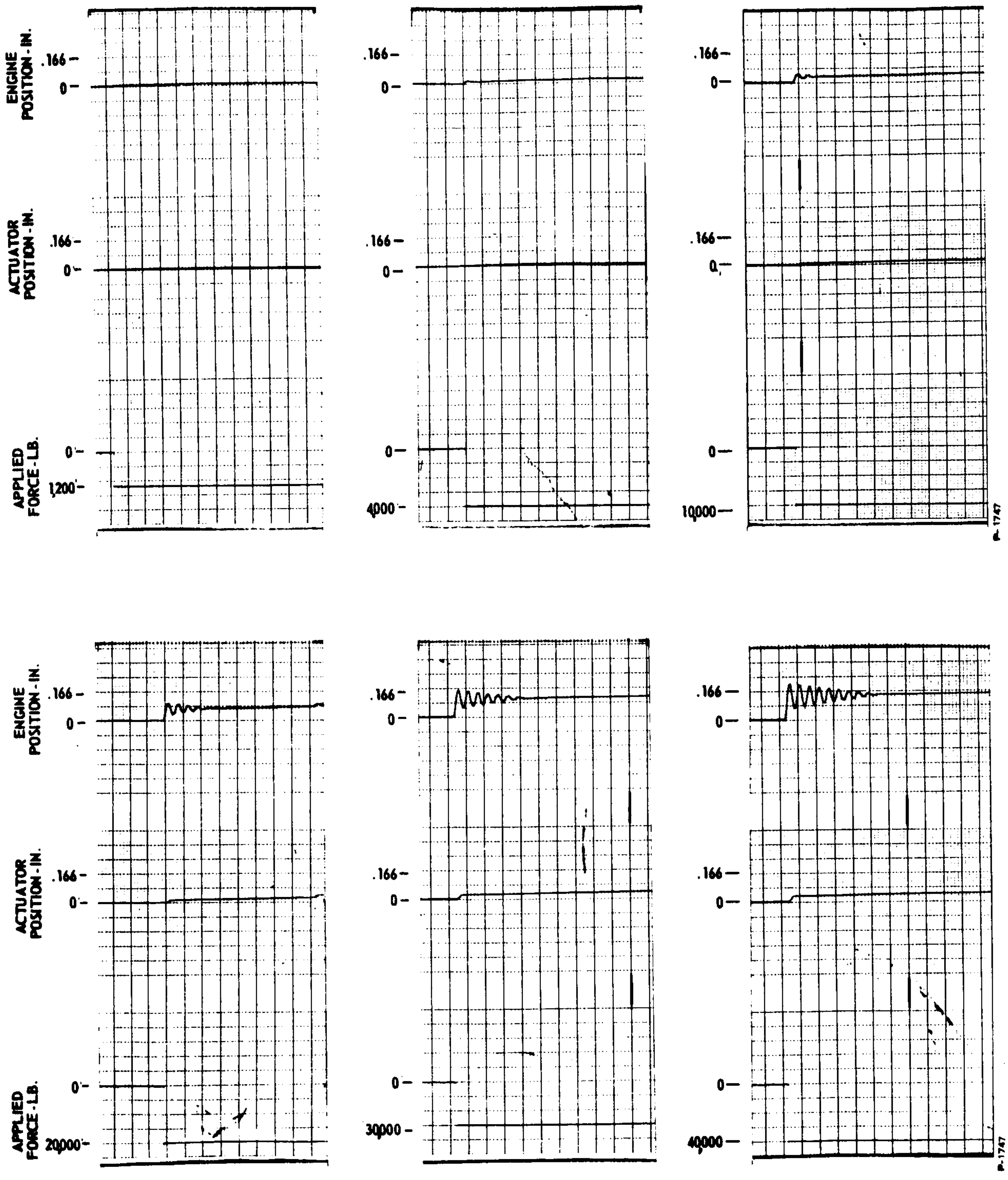


Figure C-7. Actuator and Engine Static Stiffness

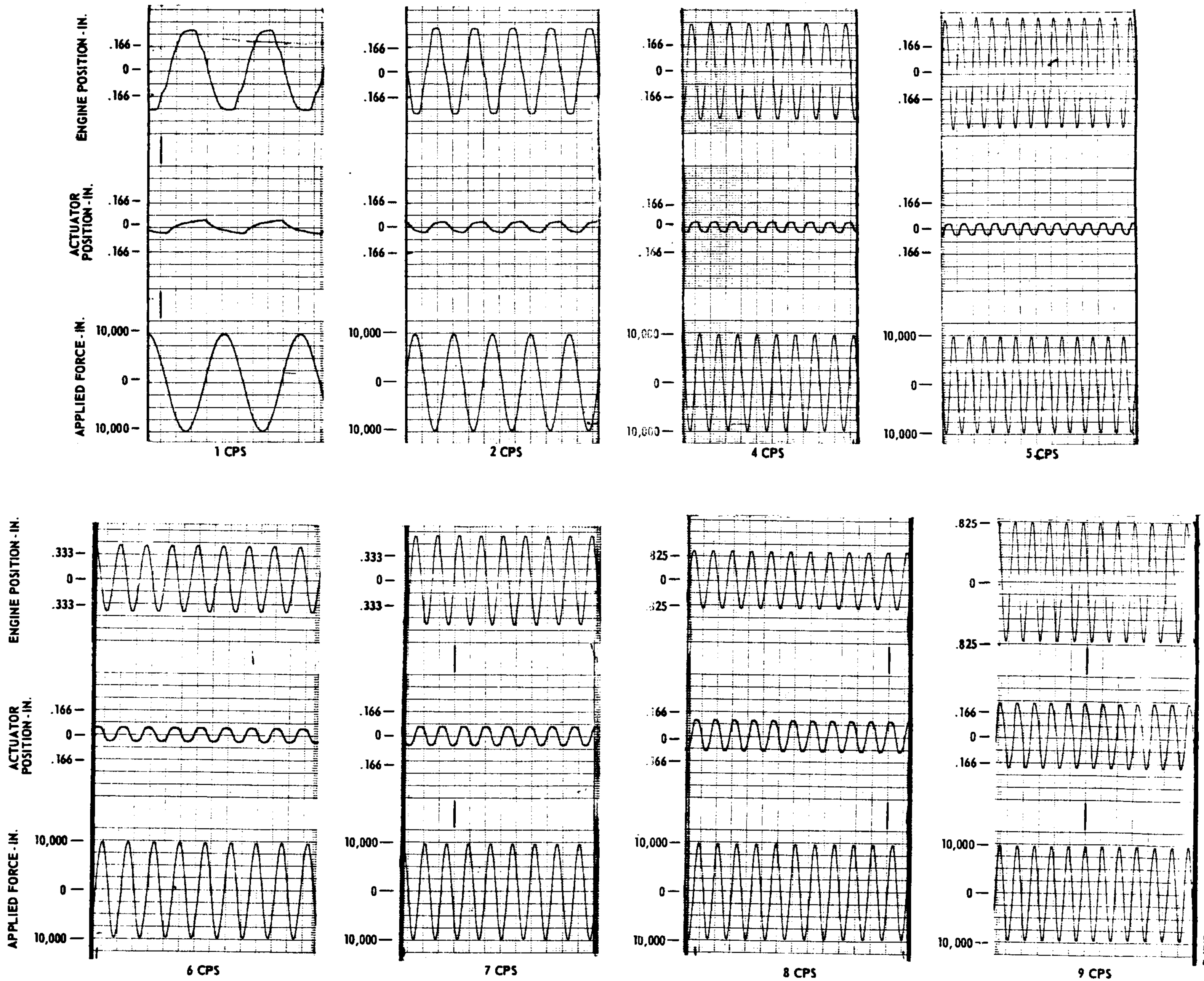


Figure C-8. Actuator and Engine Dynamic Stiffness Response

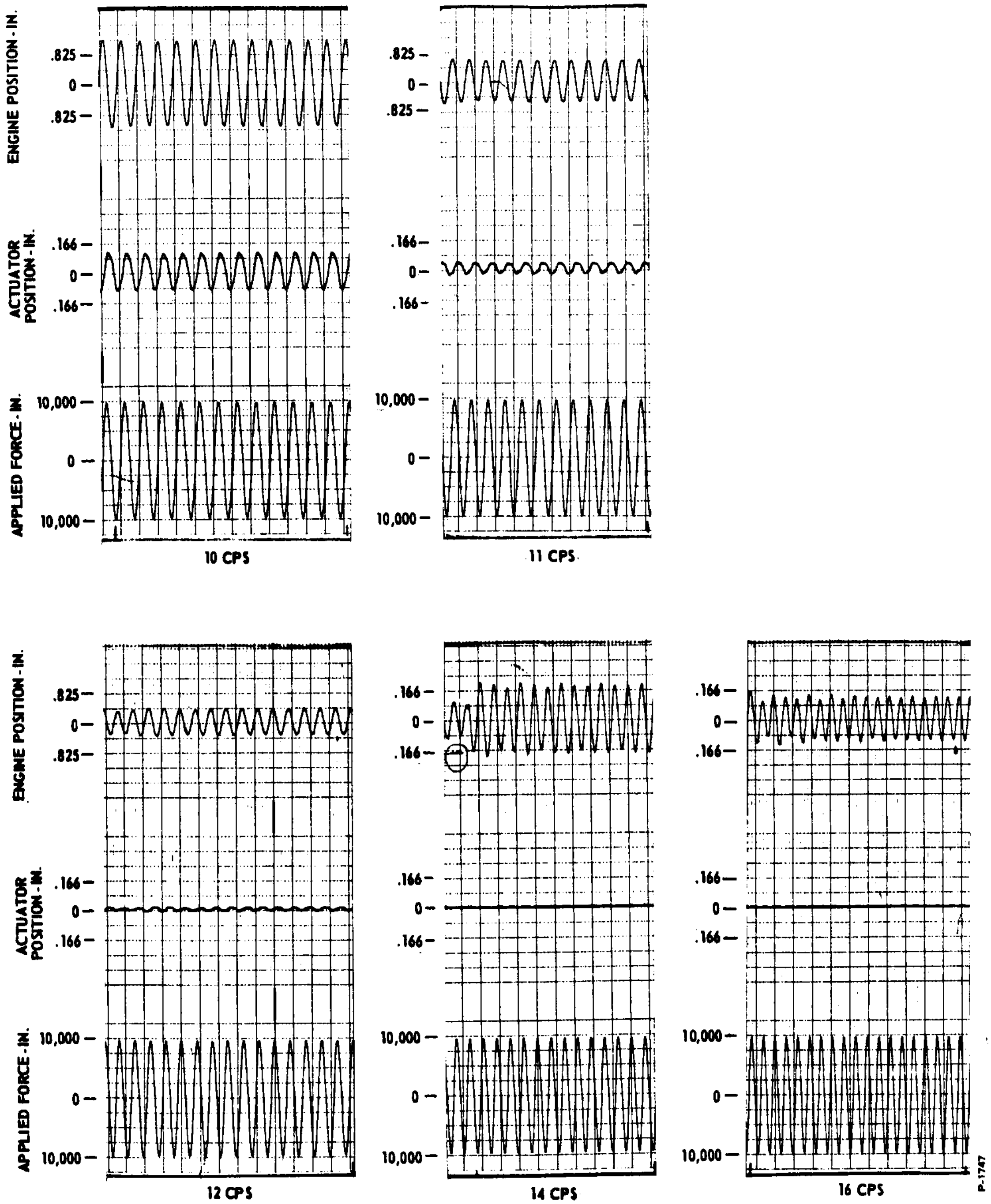


Figure C-8 (Cont'd). Actuator and Engine Dynamic Stiffness Response

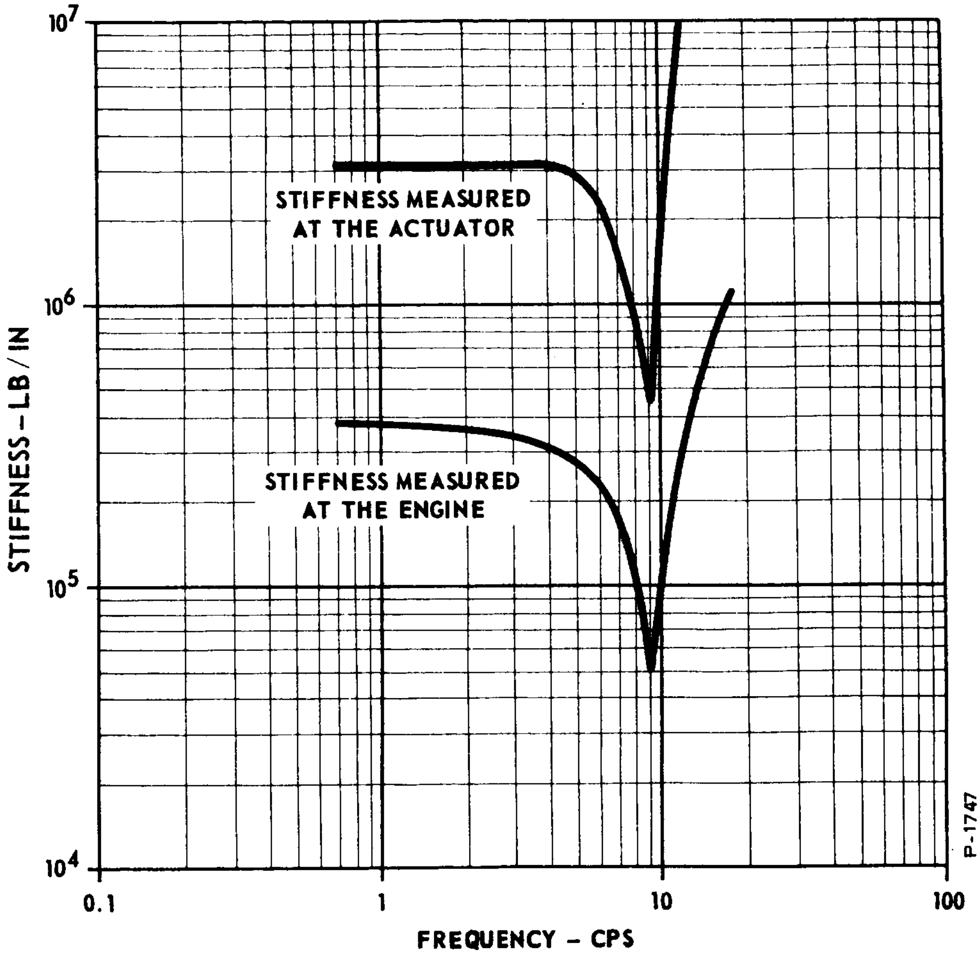


Figure C-9. Actuator and Engine Dynamic Stiffness

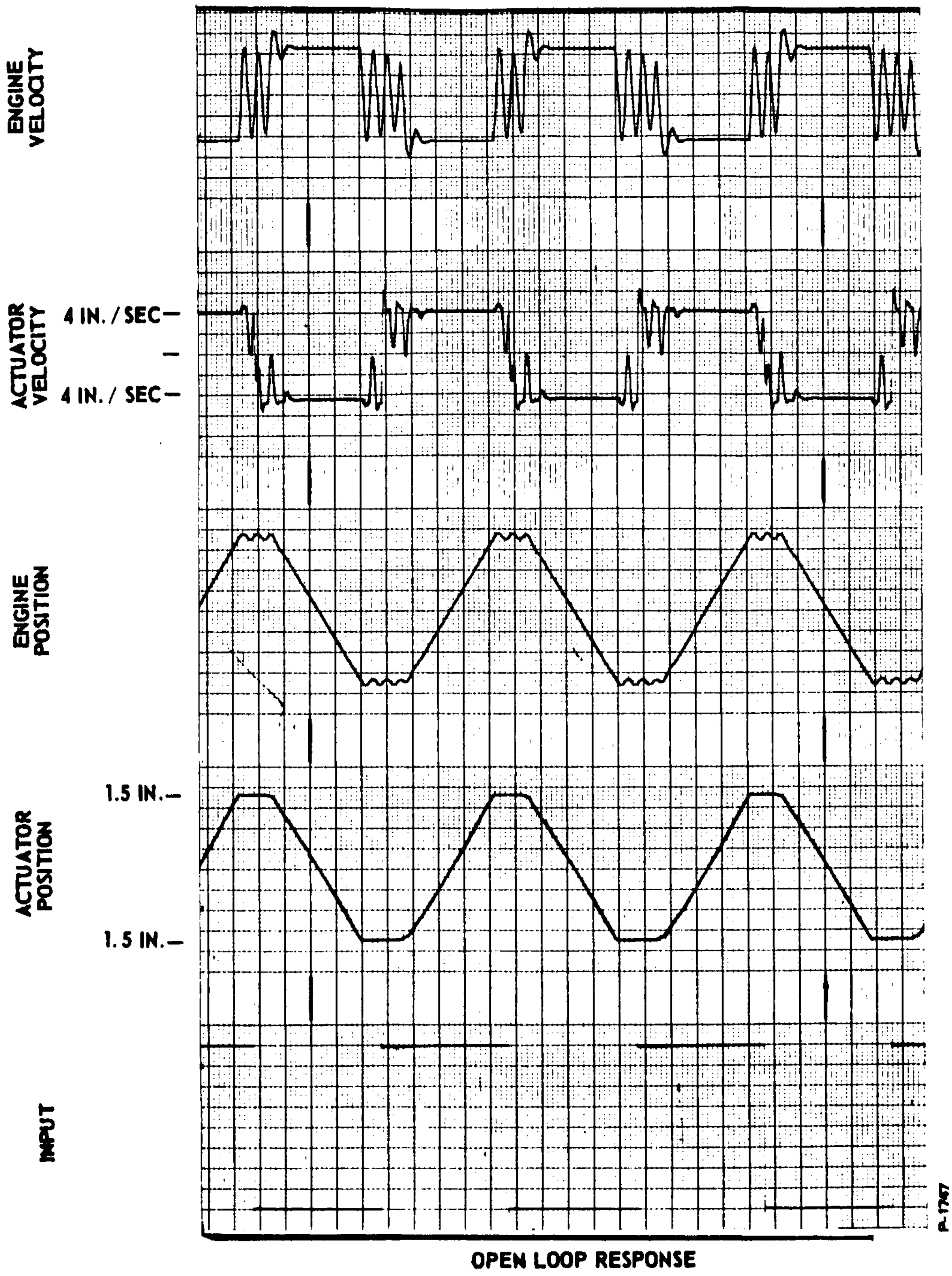


Figure C-10. Actuator and Engine Open Loop Response

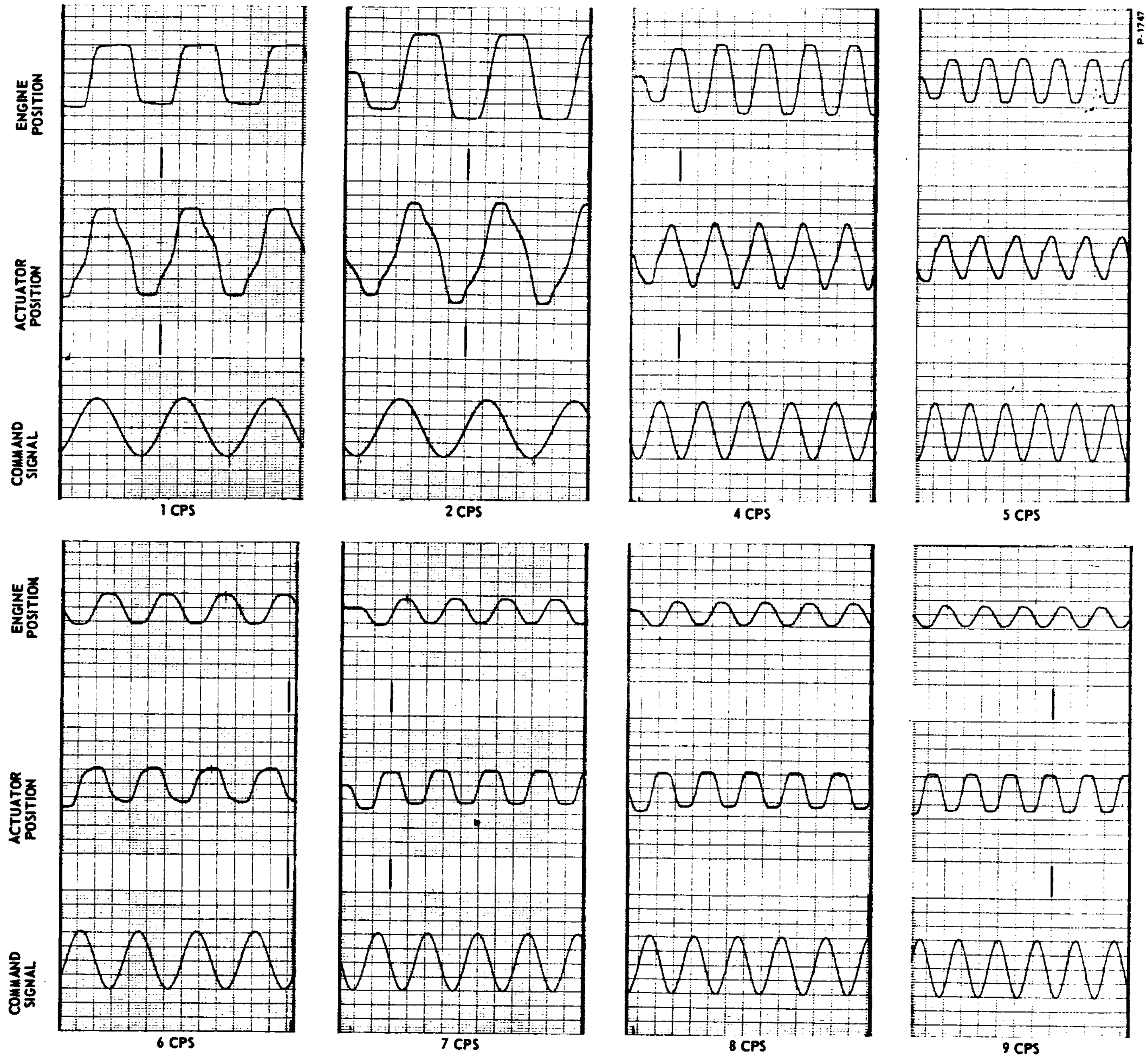


Figure C-11. Actuator and Engine Frequency Response (7300 LB Friction Load)

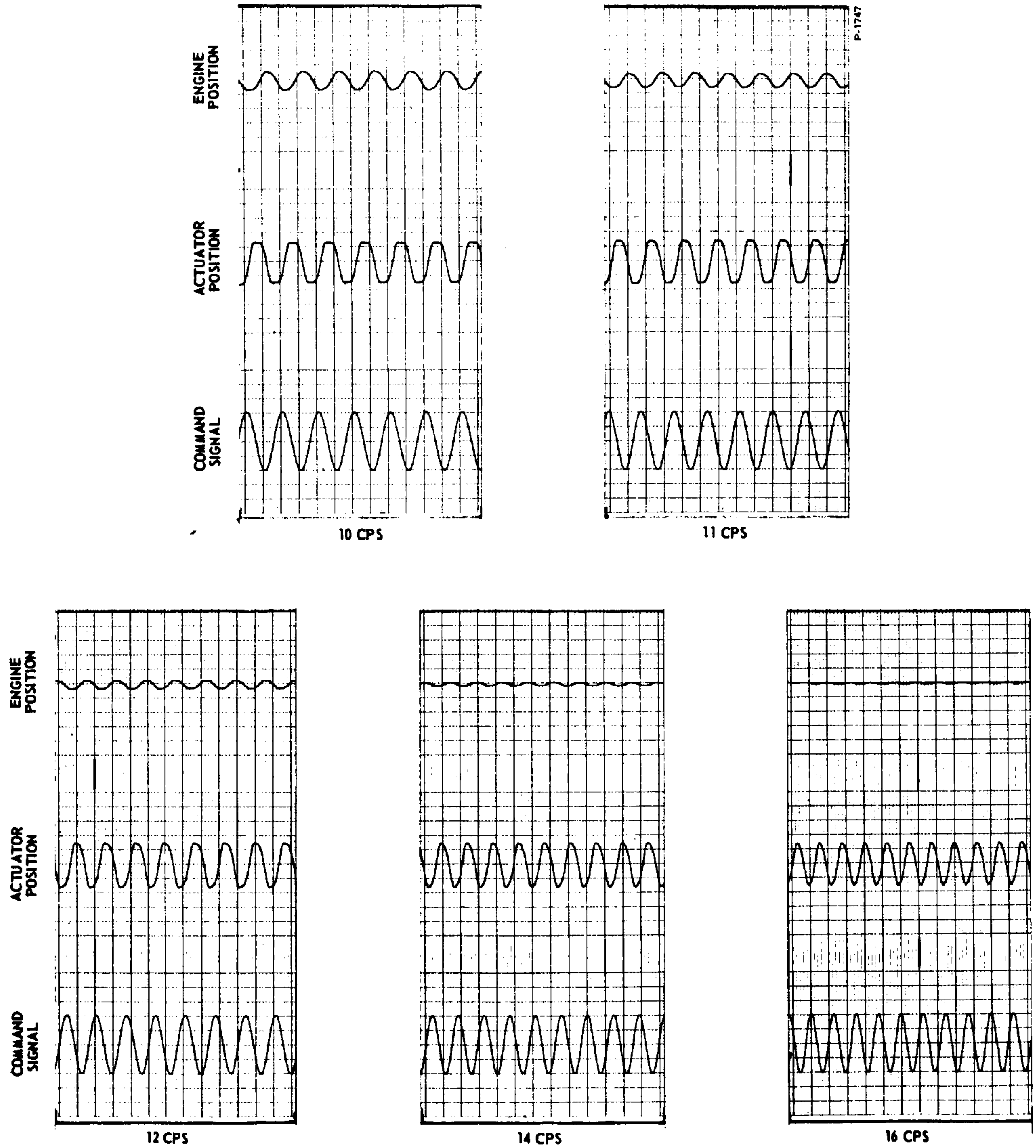


Figure C-11 (Cont'd). Actuator and Engine Frequency Response
(7300 LB Friction Load)

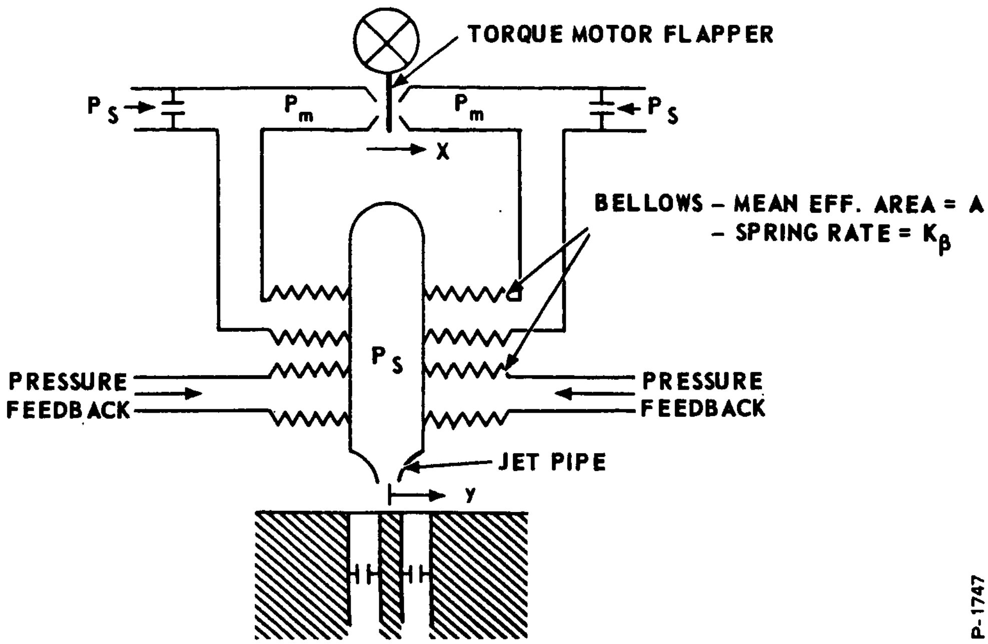


Figure C-12 - Jet Pipe Valve Schematic

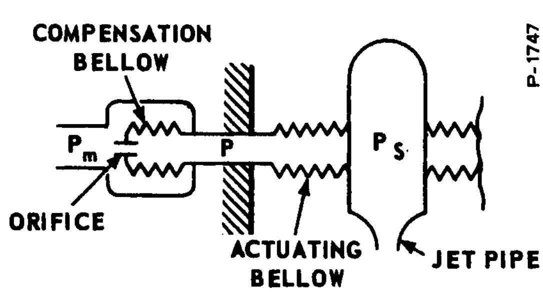


Figure C-13. Mechanical Lag Lead Network